Engineering
Library

MAR 18 1935

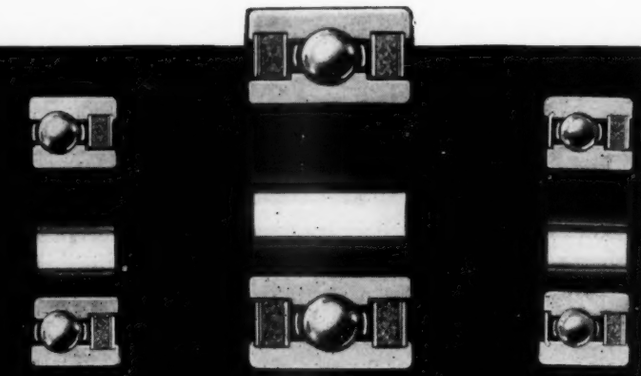
PRODUCT ENGINEERING

FEBRUARY
1935

Engineering of the highest quality and an aggressive policy in the adoption of materials and constructions best for the purpose have made of the automobile industry a leader that is showing the way out of the depression

Developments That Resulted in the 1935 Models

McGRAW-HILL PUBLISHING COMPANY, INC.



SELF-SEALED

Out of their experience of more than 23 years, NORMA-HOFFMANN engineers — pioneers in the design of enclosed and self-protected bearings —have developed and patented a range of types which meets practically every bearing condition and which affords the mechanical world

The Most Complete Line of SELF-PROTECTED BEARINGS in America

The "GREASEAL" Series of Felt-Protected Ball Bearings—in the three types illustrated above—is marked by the following outstanding features which make for better performance and more lasting satisfaction:- thick, closely-fitting felts between removable plates forming an effective labyrinth against the recessed inner ring - - - FELT SEAL REMOVABLE in its entirety for inspection, cleaning or renewal of grease - - - wide, solid inner and outer rings, with maximum contact on shaft and housing, make inserts in housing unnecessary and militate against slippage, looseness, and escape of lubricant past outer ring - - - felt seal within confines of both rings and not exposed to injury - - - seal construction avoids race distortion, assuring dimensional exactness and quiet running - - - grease capacity ample for long periods of service. . . . . Eight other types of Self-Protected PRECISION Bearings are here pictured und indexed. Write for the complete Catalog. Let our engineers aid you in selection and application.

NORMA-HOFFMANN BEARINGS CORPN.

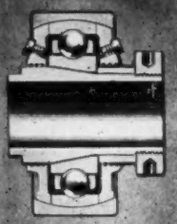
STAMFORD, CONN., U. S. A.

Type P—Closed type ball bearing with one grease - retaining plate.

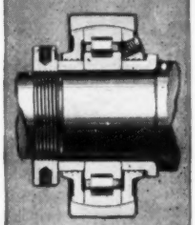
Type PP—closed type ball bearing with two grease - retaining plates.

Type GS—"Greaseal" felt - protected, grease - packed ball bearing.

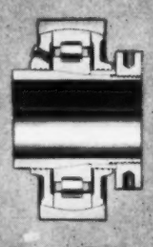
Type EGS—Adjustable, felt - protected, grease - packed ball bearing.



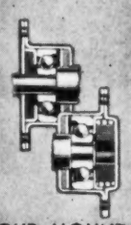
Type LTW—Self - aligning ball bearing with swiveling grease - retaining side plates and adapter sleeve.



Type RLSW—Self - aligning roller bearing with swiveling grease - retaining side plates.



Type RLTW—Self - aligning roller bearing with swiveling grease - retaining side plates and adapter sleeve.



CUP MOUNTING—Designed to use with small open type ball bearings in high-speed devices; furnished as complete units.

"NORMA-HOFFMANN"

PRECISION BEARINGS

BALL, ROLLER AND THRUST

PRODUCT ENGINEERING

VOLUME 6 NUMBER 2

MASON BRITTON, *Vice-President*

K. H. CONDIT, *Editor*
G. F. NORDENHOLT, *Managing Editor*
R. W. CARSON, *Assistant Editor*
F. A. FIRNHABER, *Assistant Editor*
LEONARD CHURCH, *Cleveland*
PAUL WOOTON, *Washington*

W. E. KENNEDY, *Manager*

Next Month

The feature article of the March number will complete the analysis of developments that made the 1935 automobiles. In this second installment will be shown how the improvements in chain drives, universal joints, clutches, thermostatic controls, rubber mountings, automobile finishes, die-castings and numerous other parts and materials made it possible for automobile designers to create the modern motor car.

Other articles will include the design of rubber bearings, the critical speed of shafts and the lubrication of ball bearings. The third article on stampings by J. K. Olsen, scheduled for February, had to be moved to the March number because of extra space devoted to our feature article.

•

McGRAW-HILL PUBLISHING CO., INC.
330 W. 42d Street, NEW YORK, N. Y.

JAMES H. McGRAW, Chairman of the Board
MALCOLM MUIR, President — MASON BRITTON, Vice-President
JAMES H. McGRAW, JR., Vice-President, Treasurer — B. R. PUTNAM, Secretary

Branch Offices: 520 North Michigan Avenue, Chicago; 883 Mission Street, San Francisco; Aldwych House, Aldwych, London, W. C. 2; Washington; Philadelphia; Cleveland; Detroit; St. Louis; Boston; Greenville, S. C.

CONTENTS for FEBRUARY, 1935

DEVELOPMENTS IN AUTOMOBILES 42
How improved materials and parts and the ingenuity of progressive designers created the modern motor car.

REDESIGNING FOR HIGH-STRENGTH STEEL SHEETS 51
V. H. LAWRENCE—A summary of the facts to be established and the things to avoid when designing with high-strength steel sheets.

LUBRICATION OF ANTI-FRICTION BEARINGS—II 54
F. LEISTER—The results of tests to determine the temperature effects of the type and quantity of grease.

LATERAL VIBRATION OF SHAFTS—I 57
LEWIS F. MOODY—A simple derivation of formulas for calculating the critical speed of a rotating shaft.

LIGHTING REQUIREMENTS 61
Fundamental data with reference to the requirements for efficient built-in lighting on machinery and other products.

INFORMATION TO GIVE WHEN ORDERING BEARINGS—II . 64
R. G. N. EVANS—The explanation of the remaining five items of the inquiry form that appeared in the January number.

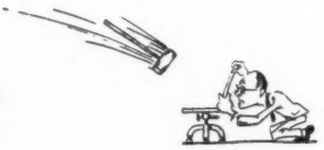
ROLLER APPLICATIONS 70
ADAM FREDERICKS—Twenty-three designs showing various methods of mounting rollers on pins.

SHORT CENTER BELT DRIVES 79
E. N. KEMLER—Formula for belt length and a chart for determining the arcs of contact for belt drives with an idler.

Editorial 41
Question and Comment 66
News 72
New Materials and Parts 74
Manufacturers' Publications 77
Books and Bulletins 78

Published monthly. Price 35c. a copy. Subscription rates—United States, Mexico, Central and South American countries $3 a year. Canada, including duty, $3.50 a year. All other countries $5 a year or 20 shillings. Entered as second-class matter January 17, 1930, at Post Office at New York, N. Y., under the act of March 3, 1879. Cable address "McGrawhill, New York." Member of A.B.P. Member of A.B.C. Copyright 1935 by McGraw-Hill Publishing Co., Inc. Printed in U.S.A. by the Schweinler Press, N. Y.

Intimate Correspondence



Help!

PRODUCT ENGINEERING
NEW YORK

I would like to obtain some reliable data or tables on co-efficient of friction between bronze 8-10-2 composition and 1040 steel. This refers to valve work when a bronze nut rotates to move a steel screw commonly Acme threaded.

Also data on the maximum pressure per area of contact between the above bronze and steel, assuming partially effective grease lubrication.

I have never been able to obtain the above information in the various handbooks but in the event you know of recent books or sources where the information may be obtained, I would appreciate it if you would so advise me.

R. C. BENSON
Camden

WE HAVEN'T the answer to this one and are printing the question in the hope that some one who knows what it is will see this and let us hear about so that we can pass along the information.

Pertinent

PRODUCT ENGINEERING
NEW YORK

The writer is preparing himself for industrial design and product development work and consequently is engaged in acquiring as much of the pertinent information on the subject as is available. He would be greatly pleased to receive reprints of the series of articles on color which appeared in *Product Engineering* in 1932 before he became a subscriber to your splendid journal. The articles are entitled:—

"An Engineer Considers Art in Engineering"—C. K. Lee—May 1932
"Color by Trial and Jury"—H. H. Adler—July 1932
"Color by Optics"—Dr. I. J. Saxl—August 1932
"Color-Materials and Behavior in Products"—F. J. Oswald—Sept. 1932
"What You Can Do With Color"—D. R. Dohner—Jan. 1932
"What Is Color?"—A. H. Munsell & Maloney, etc.—Feb. 1932
"Color—Where and How"—March 1932
"Color—Where and How"—April 1932
"Color-Materials and Behavior"—F. G. Oswald—Oct. 1932
"Effects of Perspiration on Finishes"—L. Roon—Dec. 1932

I shall be indebted to you for your courtesy in supplying me with the above articles if they are available.

BENJAMIN S. MALIN
Providence

THE only reason for printing this letter is to show what a good job on color *Product Engineering* has done. The articles in this list total 29 pages—not bad, if we do say so.

More Color

PRODUCT ENGINEERING
NEW YORK

Please let us have some information regarding color schemes. Some time ago a series of articles appeared in *Product Engineering* on this subject.

The reason for this request is that we expect to erect a service station soon for specialized automotive service work—brakes and wheel aligning. We would like suggestions regarding colors to paint equipment, walls, floors, etc.

We shall appreciate your cooperation.

G. EARL KOCH
Cincinnati

WE ANSWERED this one by referring the writer to several competent experts on the subject of color. As a matter of fact this particular application of color principles is just a bit beyond our field although we are interested, of course, in the colors of the products involved.

Usable Curves

PRODUCT ENGINEERING
NEW YORK

At last the obscure atmosphere surrounding plain bearing design begins to clear and a really practical article appears in the December issue of *Product Engineering*. Victor Tatarinoff is to be congratulated for reducing the hitherto involved equations to a series of usable curves based upon A.S.A. standard running fits.

FREDERICK FRANZ
New Haven

IT TAKES a good man to reduce curves of this sort to practical usefulness. But there are other curves that are intrinsically usable to any of us, if you get what we mean.

Exploited

PRODUCT ENGINEERING
NEW YORK

It is refreshing indeed to read such articles, as the one written by Mr. R. Wosak and appearing on page 2 of the December issue, under the heading of "Good Designers," as also your comments.

Such executives, as the chief engineer referred to by Mr. Wosak, are even scarcer than good designers, and he deserves to find the best men for his business.

What I would like to emphasize here is the fact that a good designer is mostly so absorbed in his problems that he forgets to think of his own advancement, with the result that more often than not he is taken advantage of to the limit, and discarded at the first opportunity.

The same can be said with regard to shop men, especially in the highly skilled classes.

It is very unfortunate, to say the least, that far too many employers, large and small, are penny-wise and pound-foolish enough to indulge in this practice of exploitation. They are the very stumbling blocks to real progress, inasmuch as they smother ambition and good will of such men who possess the essential tools for progressive development, i.e., imagination and sound experience.

A. CASAGRANDE
Upper Darby

WE DIDN'T set out to make this page a place for carrying on such discussions as this—they really belong on the editorial page—but since we seem to have started something when we printed Mr. Wosak's comments, it is evidently only fair to continue it to a finish. Further comment from us would probably be superfluous.

Three Bites

PRODUCT ENGINEERING
NEW YORK

On page 26 of the January issue of *Product Engineering* is an article entitled "Sawing with an Electric Arc," by Mr. M. C. Smith.

Please advise the name of the company manufacturing this equipment for sawing, machining, etc., by means of the electric arc, as we are interested in their line and would like to receive catalogs and literature describing it.

A. H. MURRAY
Rochester

THIS is one of three similar letters that came in the same mail, one from New York State, one from Pennsylvania, one from Kansas. And there will probably be others because these arrived shortly after the issue went out. Our best answer to those who have written, and to those who want to know but haven't written is to refer them to the author, M. C. Smith, 6075 Franklin Ave., Hollywood, Calif.

Another Repeater

PRODUCT ENGINEERING
NEW YORK

We observed in the *Product Engineering* of December 1934 on page 448 a photograph of a "Battery-less flashlight."

We would like to obtain more information on this device such as descriptive material, price, etc. If you are unable to furnish us with this information will you please advise us where we can obtain it?

O. R. WYSONG
Detroit

TO SAVE time and energy, both for ourselves and for others who want to know where to get this gadget—quite a few have already written about it—let us admit right off that this flashlight is a European product, but that it is sold in this country by Alex. Taylor & Co., 22 East 42nd St., New York City. The retail price is around $3.

PRODUCT ENGINEERING

KENNETH H. CONDIT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. FEBRUARY 1935 VOLUME SIX, NUMBER TWO

The Time Is Ripe

IN HIS MESSAGE *to stockholders, employees and the public, relative to the impending consolidation of Republic Steel, Corrigan-McKinley and The Truscon Steel Companies, T. M. Girdler, president of the Republic Steel Corporation, presented his opinions as to the immediate future of American industries. The following was abstracted verbatum from that message. It expresses, better than we could, the significance of the changes now taking place.*

IT IS difficult to recognize the birth of a new period, even though we stand in its presence. The reason is the human tendency to resist change. New ideas and new products are under suspicion until suddenly it becomes apparent that these new things are supplanting the old. When this is recognized, there follows a sharp, widespread awakening of interest, and a general move forward to participate in it. Such a change is now occurring. An impulse has already started that is bound to be recognized soon as the beginning of another giant move forward.

You may recall how automobiles were ridiculed until the public could no longer withstand the pressure of this new idea. Suddenly, all rushed to adopt it. And our greatest modern industry was born.

The public regarded with great skepticism the earlier construction of skyscrapers, modern bridges, airplanes, stream-lined trains, air-conditioning, electric refrigerating—and hundreds of similar products of modern engineering. It was not until the pressure of these new ideas became too strong for doubt, that they were recognized as practical realities. Those who first shared in the forward movement, profited most.

STREAM-LINED trains, such as the "Zephyr" type, have caught the public imagination by their amazing performance. These trains are no longer experiments. They provide practical transportation, and the railroads are adopting them. They could not have been built ten years ago, because their efficiency is not just the result of stream-lined design. It is due to the fact that new metals, notably stainless steel, effectively meet the new requirement for greatly increased strength combined with lightness. Such metals simply could not be fabricated only a few years back.

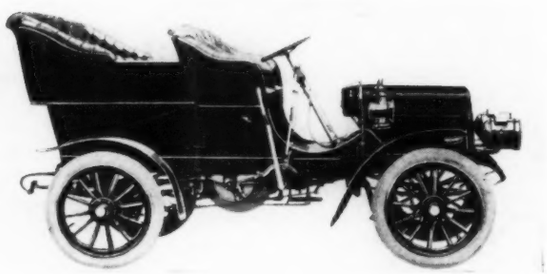
I cannot put any limit on the demand for new metals. We know that vast replacement of old-fashioned metals and materials must soon be made. New inventions are creating a demand for new metals that ought to reach enormous proportions.

I am convinced by every evidence that America is again moving forward into a new cycle of greater achievement and greater production. We know there is at hand a deep stirring presaging another great industrial development.

The time is ripe to begin a vigorous move forward.

Developments That Made The Motor

In the design of all products the ultimate is limited only by the qualities and properties of the materials available. Automobile designers are noted for designing up to that limit and keeping abreast with the improvements in materials and manufacturing processes. They have often lead the way in developments that later found general adoption in the design of products in other fields.



1904

NO OTHER consumer product involves a greater range of problems than the automobile. And because problems in the design of almost any mechanical product find their counterpart in the modern motor car, engineers in all fields of product design realize that the progress of the automotive industry materially influences their own design problems.

Advances in automobile design naturally fall in two major divisions. In one division are all those developments that have to do with the motor car as a vehicle for carrying the power plant, operating equipment and passengers. Body design, chassis frame, seating arrangement, trim, finish, riding qualities and appearance can all be classified as pertaining to the automobile as a vehicle. Developments in the power plant, transmission, and controls, form the second major division.

Body styles and appearance have changed fundamentally as the accompanying pictures demonstrate. From the "open buggy" of 1904 successive developments over the past three decades have brought the streamlined motor car of today. Behind these pictures, however, lies a story of new processes, improved materials and better types of construction that have made it possible for body designers to achieve progress.

When in 1912 Edward G. Budd conceived the "all-steel" spot welded body, it marked the beginning of a new development which has had its counterpart in many fields of design. These first all-steel bodies were assembled by spot welding and the seams finished by gas welding, with a few riveted connections.

But in the early days of the steel body, limitations imposed by available presses, die equipment and steel sheets greatly restricted body design. Many desirable structural features had to wait until spot welding and gas welding machines and methods were improved; until new rolling mill equipment was installed to roll larger sheets; and until metallurgical control and more attention to finish gave sheets that could withstand more severe forming operations.

Although the desirability of the closed car had always been apparent it has only been within the last nine years that these new developments have made possible the design of closed bodies for cars in the low-price class. But even with new body-building equipment and better sheet steel it was still necessary to use clinched seams to a considerable extent.

Recently, the development of flash welding, the resistance welding of sheets edge-to-edge, in combination with other improvements in welding methods and equipment has resulted in low-cost, strong and rigid closed bodies made entirely of steel. Tool and equipment costs, however, have generally limited the all-steel bodies to the high production models.

Simple as all this sounds, many difficulties had to be overcome. The development of deep-drawing sheet steel not subject to "stretcher strains," especially in the larger sizes of sheets, and steels that would withstand the severe forming operations, required a great deal of metallurgical research and the experimental development of a new technique in die design and forming

Cars of 1935

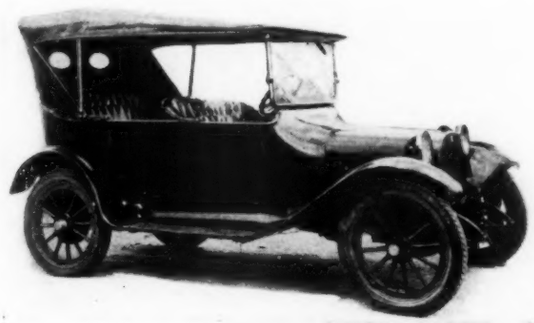
operations. With these problems solved, the automobile designers were free to go ahead with designs that called for entire sides stamped out of a single sheet, with panel relief and stamped beads for decoration, with skirted fenders and streamlined contours.

The all-steel roof in some of the new models is the latest development in body design. In this construction, a single stamping replaces the wire, fabric and felt combination formerly used, making the closed car literally all-steel. Sound deadening the steel roof sheet was not a difficult problem. To eliminate drumming, door and body panels for several years have had a jute fiber blanket cemented to the sheet by an asphaltum mixture. This method of sound deadening is used for the all-steel roof. The two factors that contributed most to making the steel roof design commercially possible were the development of presses of sufficient capacity and size and the extension of the flash welding process. New die manufacturing equipment played an important part in reducing the cost of the dies required to produce the difficult roof contours.

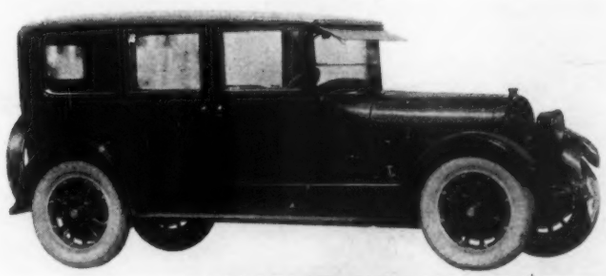
In quantity production, the steel roof is cheaper than the wire and fabric construction. But in addition to this, its real superiority lies in the added safety that it affords, a factor of increasing importance because of the demands for greater safety in conjunction with higher car speeds. This feature is in the same category as safety glass for windshields and windows, which is now a legal requirement for new cars in several states and is becoming standard equipment throughout the automobile industry.

Increasing speeds also served to accentuate the shortcomings in the riding qualities of automobiles, particularly at the rear seat. But it was not until years of research had perfected the spring materials and spring design that the independent wheel springing with helical springs became possible. The most important factor in development of independent front wheel suspensions was the desire to obtain a low spring rate which could be accomplished most readily by helical springs. But one of the problems in this type of design was to procure a material that would not fail from fatigue.

To avoid fatigue failure, the helical springs for the front wheel suspension are coiled from bars that have been ground on centerless grinders. The ground surface finish is necessary to prevent fatigue failure resulting from cracks that have their beginning at some surface imperfection. Most of these springs are made



1914

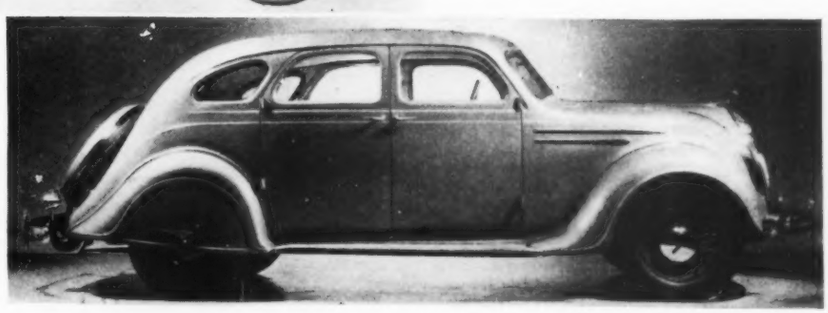


1925

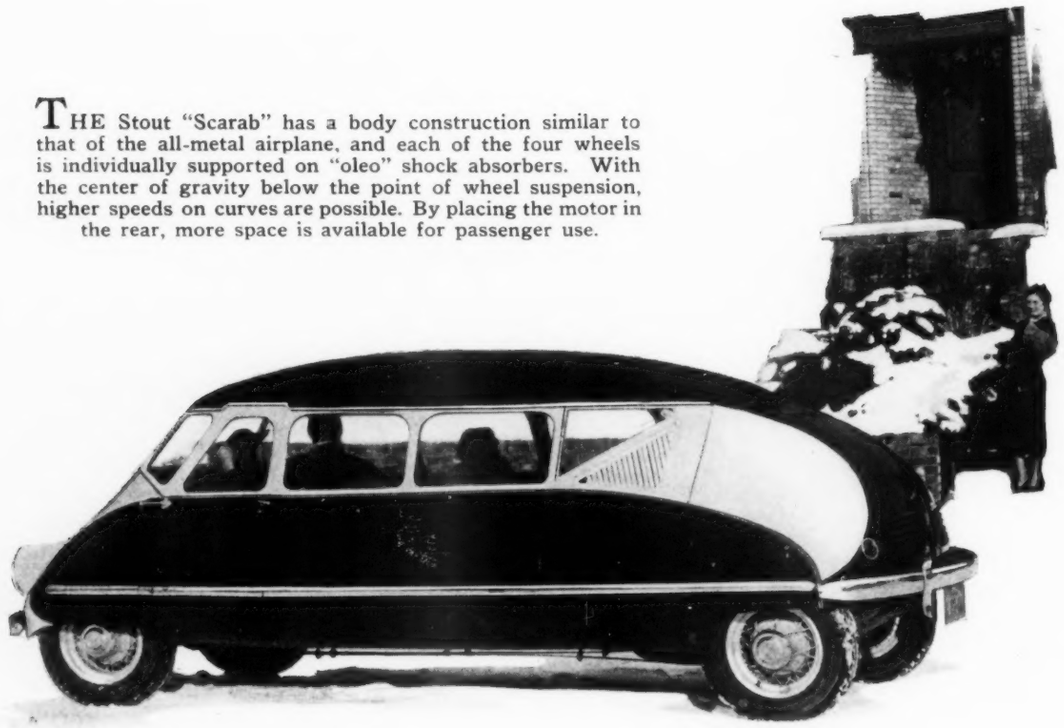
Until modern welding methods were developed, body designs were limited to open models. The period from 1904 to 1914 witnessed the introduction of electric headlights and starter

New manufacturing methods and better materials and finishes made possible the engineering improvements embodied in the 1935 models

1935



THE Stout "Scarab" has a body construction similar to that of the all-metal airplane, and each of the four wheels is individually supported on "oleo" shock absorbers. With the center of gravity below the point of wheel suspension, higher speeds on curves are possible. By placing the motor in the rear, more space is available for passenger use.



of silicon-manganese steel and are heat-treated in a reducing atmosphere to avoid scaling.

Because some form of vibration-damping device is used in connection with the independent front wheel suspension there is little or no possibility of spring flutter such as has been experienced in the valve springs of high-speed engines. These vibration-damping devices for the front wheels are equivalent in principle of operation to the familiar hydraulic type of shock absorber.

Riding comfort in a number of 1935 models has been improved by a better weight distribution between the front and rear axle. By moving the motor forward over the front axle, the added weight has permitted the use of a higher spring rate, in addition to making available more room in the body. By shifting the rear seat forward of the rear axle, riding comfort in the rear seat is better and it allows increased space in the rear for baggage or a pocket for the spare tire.

It is only one more step to utilize the baggage and spare-tire space in the rear as an "engine compartment," thereby achieving the much talked of rear-engine car. This is what actually has been done in the car designed by William B. Stout. Although only samples of the Stout car have been made, the design warrants serious attention. In the opinion of many, some of its features forecast the automobile of the future, not only because of the position of the engine, but also because of the new type of body structure and other advanced features incorporated in it.

In the Stout car, body and chassis are combined in one structure built upon a frame consisting of a series of vertical hoops and longitudinal members—a type of construction characteristic of all-metal airplanes. Alloy steel tubing is welded to form the framework of the car, with the design based on stress analysis including crash strength as well as actual strength. In some respects the principles of body and chassis construction are similar to those found in the Adler-Trumpf body described in *P.E.*, June, 1934.

Space occupied by running boards, hood and front fenders in conventional cars is inclosed within the body of the Stout car, and is available for passenger use. Only the driver's seat is fixed to the car; the other seats can be moved around as desired. This arrangement of movable seats is permissible, it is claimed, by virtue of the exceptionally smooth riding resulting from a new type of wheel suspension, in which each wheel is independently hung on an airplane-type shock absorber from a point above the center of gravity of the car. Thus, in rounding sharp corners at high speed the car tends to tilt toward the inside of the curve, the wheel action being analogous to the action of the swing-bolster trucks long used on railroad cars.

Higher Engine Speeds Bring Forth New Problems

Alloy cast iron for cylinder blocks and aluminum for pistons used to reduce wear and inertia forces

Because the engine is the heart of the automobile, it is not surprising to find that more fundamental improvements and sweeping changes have been made in the design of the motor than in any other part. Public demand made necessary ever greater horsepower to give higher car speeds. To obtain this increased horsepower without going to bigger engines, it was necessary to increase engine speeds. At the same time, to

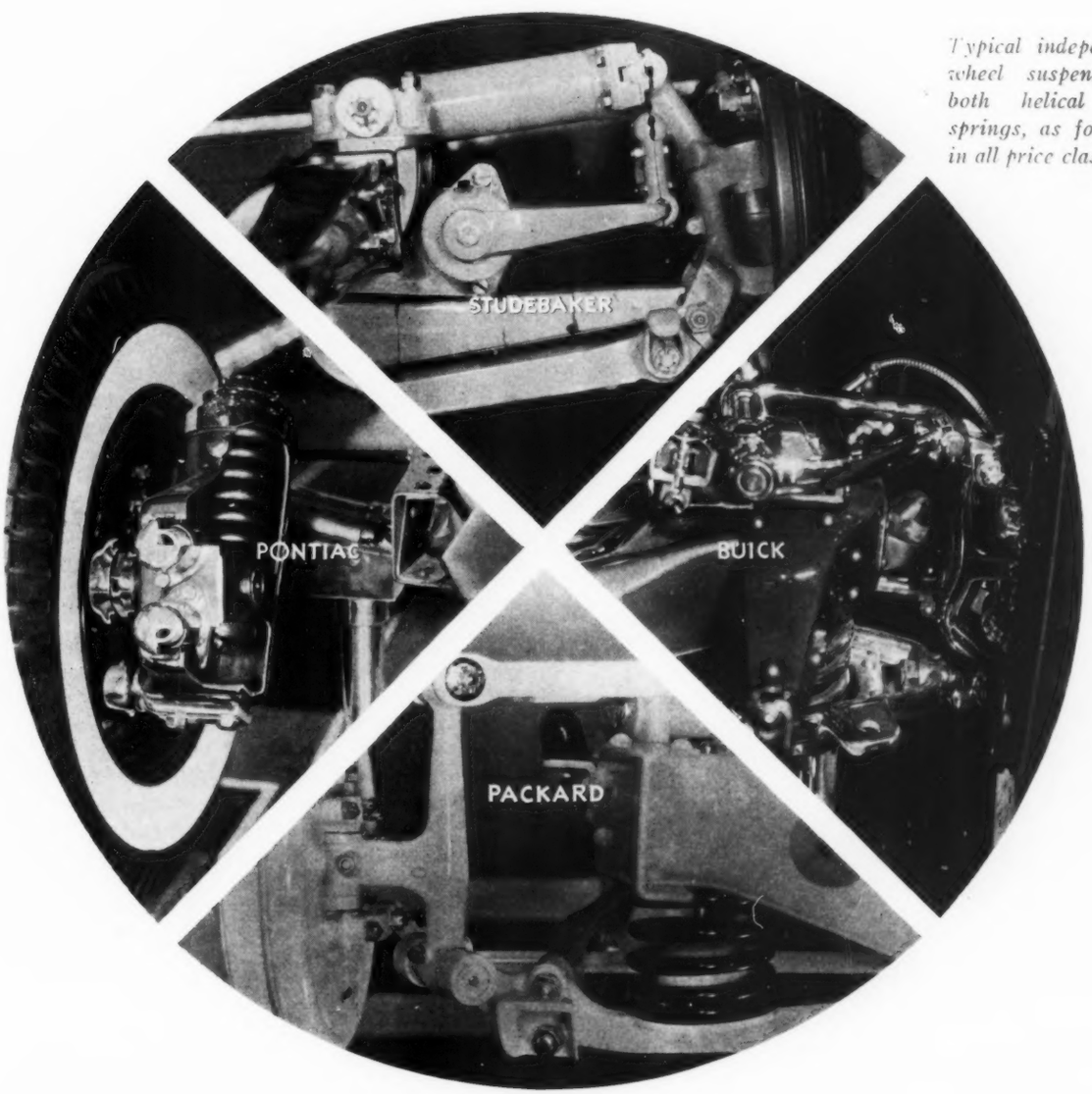
maintain or improve engine performance, it was necessary to go to higher compression ratios, a factor also influenced by the fuel characteristics. But the small, powerful, high-speed engine aggravated the always-existing problems of valve and valve seat wear, valve cooling, cylinder wear, lubrication, bearing life and vibrations.

The problem of eliminating the extremely rapid wear in the cylinders caused by the increased speed presented two possibilities. One was the use of liners; the other was to use a harder and more-resistant material for the cylinder block. The development of Cr-Ni and Cr-Ni-Mo alloy cast irons made available a material far superior to plain gray iron in that these alloy cast irons are hard, dense and highly wear resisting. But being harder, the material is more difficult to machine. It was not until the development of diamond cutting tools and the new tungsten-carbide materials made possible the economical machining of these harder and tougher alloy cast irons, that they came into general use. But even with these high-speed cutting tools, the maximum alloy content with its accompanying hardness is limited by the machineability of the material.

One manufacturer of industrial engines first used a nickel cast iron, containing 1.5 per cent nickel, in 1924, in engines for tractors and combines operated in areas affected by wheat rust. Because of the abrasive action of the dust, ordinary cast-iron bores would be worn out in one season, while the alloy cast iron showed no more wear than found under normal service conditions. With increasing speeds and higher operating temperatures, the nickel cast iron was adopted as standard for all truck engines, and two years later was used for passenger car motors. In addition to giving longer wear, a much higher surface finish can be obtained with little added machining cost, greatly reducing the "running-in" mileage.

Composition of Alloy Cast Irons

For the nickel-chromium alloy cast irons the nickel ranges between 2½ and 3 per cent, with not more than ½ per cent chromium. In the Cr-Ni-Mo cast irons, molybdenum usually ranges between 0.50 and 0.60 per cent, with nickel up to 0.30 per cent, although for



Typical independent front wheel suspensions using both helical and leaf springs, as found in cars in all price classes

cylinder liners the molybdenum may run as high as 1.00. For truck engines the specifications adopted by General Motors Truck Corporation are:

	Per Cent
Total Carbon	3.10-2.40
Combined Carbon	0.75-0.90
Silicon	1.90-2.10
Manganese	0.55-0.75
Phosphorus	Max. 0.20
Sulphur	Max. 0.10
Nickel	1.80-2.20
Chromium	0.55-0.75

Heat Treatment: Cylinder liner castings with a wall thickness not greater than $\frac{3}{8}$ in. shall, after being heated in an oven-type furnace at 1,540-1,560 deg. F. for 30 to 40 min. and quenched in still oil, have a hardness of not less than Brinell 512 or Rockwell C-52.

Hardness (as cast)—Brinell 212-241.

Transverse Strength (A.S.T.M. arbitration bars)—2,400 lb. minimum.

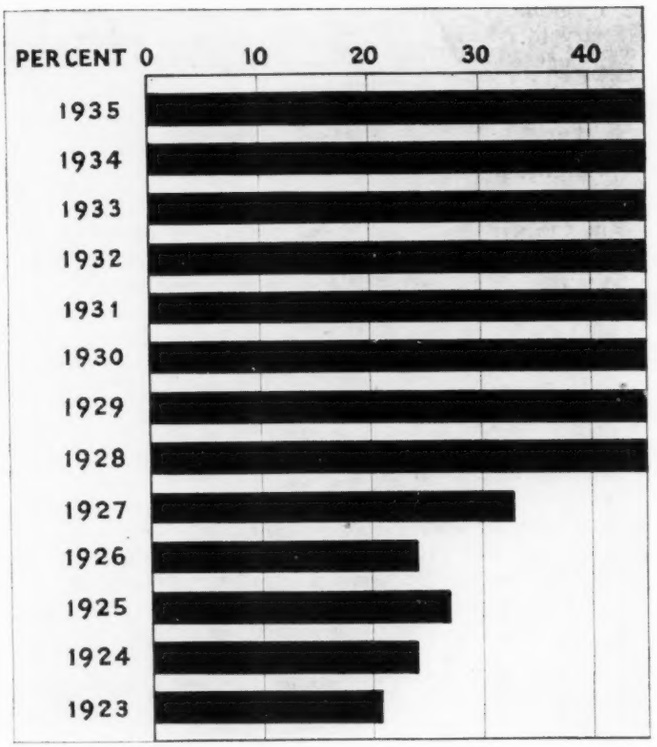
Transverse Deflection—0.20-0.30 in.

Tensile Strength (test bars machined from castings)—37,000 lb. per sq.in. minimum.

Better cooling in the cylinder head offers another possible method of increasing performance through the use of higher compression ratios. Cylinder-head material with a higher coefficient of heat conduction eliminates hot spots and pre-ignition. Composite heads with aluminum combustion chambers inserted in a cast iron head have been used to a limited extent, and experiments are being made with cast copper heads. But the greatest progress in this direction has been in the use of a cast aluminum cylinder head. In 1933 two makes of cars appeared with cast aluminum cylinder heads. Among the 1934 makes of cars, twelve were equipped with alumium cylinder heads, with the same number of manufacturers using aluminum cylinder heads in 1935 models.

In the opinion of some automobile designers, the weight saving through the use of aluminum cylinder heads is of far greater importance than the improved heat conduction obtained. As much as 20 lb. saving in weight is possible.

It is really immaterial which advantage is the more important, improved cooling and combustion, or saving in weight. The fact is that among the twelve makes of cars using aluminum cylinder heads in the 1935 models are found nine makes selling for less than $1,000: Chrysler, De Soto, Dodge, Ford, Graham, Hudson, Lafayette, Reo, Studebaker, and Terraplane.



The trend to higher motor speeds has impelled the more general adoption of aluminum pistons to keep down inertia forces

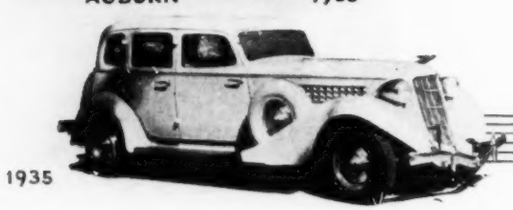
Less Friction With Tinned Pistons

With the increased engine speeds and higher compression pressures, it was necessary to improve the lubrication and wear resistance in the cylinder bore and on the piston, particularly during the breaking-in period. Unless some provision is made to permit higher engine speeds during the breaking-in period, the car speed during "run-in" would be excessively low, as measured by present-day standards.

In the cars using cast iron pistons, more satisfactory "breaking-in" speed is made permissible by tinning the piston. The soft tin coating betwen the cast-iron surfaces of the cylinder and piston flows into the low spots, rapidly producing a mirror surface on both piston and cylinder bore during the initial period of use. The soft tin also acts as a lubricant, decreasing the frictional drag.

For greater strength and higher machineable hardness several companies are now using a cast-iron piston with a total carbon content of 3.25 per cent, silicon at 2.25 per cent, and 0.35 per cent molybdenum. A

AUBURN 1928


1935

Hudson rings are pinned to prevent rotation in grooves

small amount of nickel, usually about 0.20 per cent, is added for increased strength. As compared with the alloy cast iron in general use for cylinder blocks in the higher price classes, this composition has a little less silicon and molybdenum. An added advantage of this alloy cast iron is better structural stability throughout the operating temperature ranges encountered in the cylinder combustion zone.

Higher motor speed has also made necessary a considerable reduction in inertia of the piston and connecting rod assembly. By the use of light-weight pistons of both cast iron and aluminum, bearing pressures and vibration forces have been reduced. And the effect of this weight reduction had its influence on many other elements in the design of the connecting rod and piston assembly.

Adoption of aluminum pistons has increased greatly during the past five years but this change has also brought with it new problems peculiar to aluminum. In the early designs it was found necessary to allow an excessive amount of piston clearance when cold in order to prevent seizing when the motor became hot. Further developments in the design of the aluminum piston have overcome the clearance difficulty through the use of various types of designs such as the invar-strut piston, and the split-skirt and T-slot types, and the development of aluminum alloys having a lower coefficient of expansion. Another disadvantage of the aluminum piston was its higher wear rate, as compared to cast iron. By the anodic treatment or surface oxidizing, aluminum pistons are now given a hard wear-resisting skin which reduces the rate of wear.

Trend to Cast Iron Camshafts. Ford Pioneers Cast Crankshaft

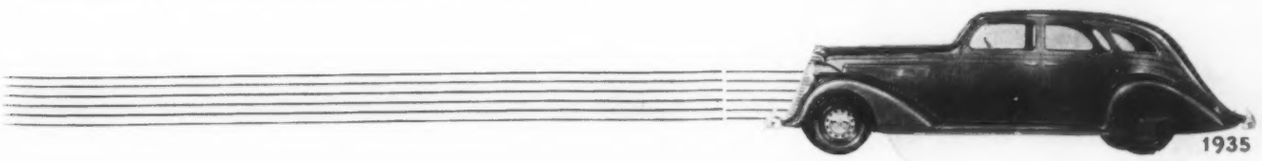
Following application on 1934 test cars, Ford now standardized on cast crankshaft of steel. Tests on alloy cast iron crankshafts continue

Reduced blow-by and better top ring sealing has been obtained by the adoption of a wider land above the top ring in aluminum pistons. This construction reduces the temperature of the top ring and thereby reduces the tendency of oil to carbonize and freeze the ring to the piston. Ribs have been added to the under side of the aluminum piston head to assist the flow of heat from the head to the skirt where it can pass through the cylinder wall to the cooling water. This goes hand in hand with extending the cooling-water cylinder jacket down to the bottom of the cylinder, a feature of a number of the 1935 models.

Recent demands for increased speeds in bus and truck engines has also led to the extensive use of aluminum pistons where formerly cast iron had been used exclusively. Although the tin-coated cast-iron piston costs less and has somewhat longer life, light-weight aluminum pistons greatly increase bearing life under high speeds now encountered in bus and truck service.

Connecting rod design has changed but little during recent years in regard to the materials used. However, it is now common practice to forge the studs integral with the rod for the purpose of reducing manufacturing costs.

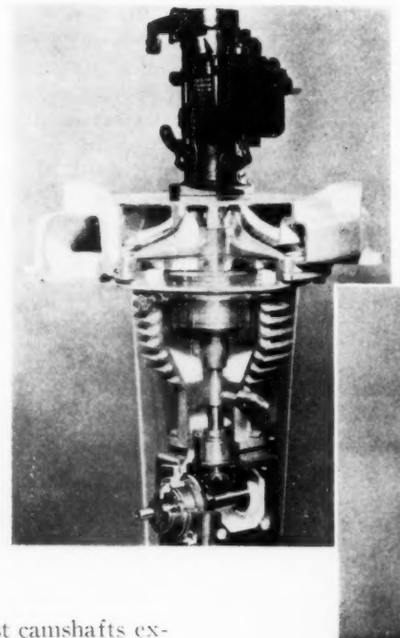
An outstanding development during the past five years has been the adoption of a cast camshaft by sev-

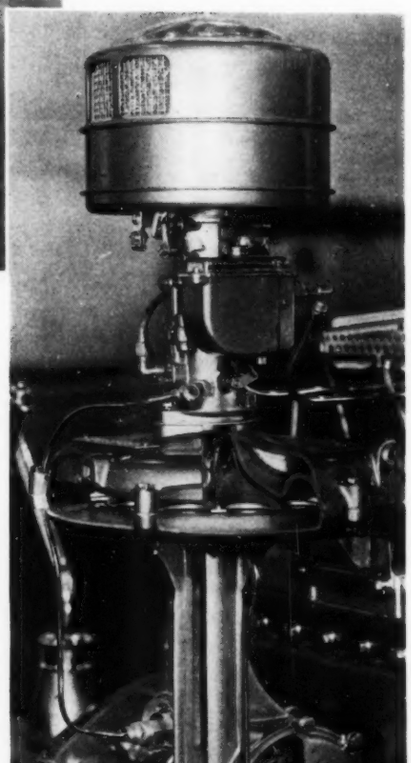


eral motor manufacturers, and the cast crankshaft now used by Ford. This is the result of metallurgical research from which developed a new method of casting, giving a shaft of the required strength and stiffness. Because of the initial expense involved in setting up the casting process, the use of cast camshafts and crankshafts has been limited to applications requiring comparatively large quantities. However, it is reported that the cast-iron crankshaft is being considered by nearly all of the automobile manufacturers.

The composition in most general use for development work on cast crankshafts is a cast iron with a total carbon content of 2.20 to 2.75 per cent, molybdenum from 0.75 to 1.50 per cent, and about 2.50 per cent silicon. This is approximately the same as the composition used for cast camshafts except for a somewhat lower molybdenum content.

Increased speeds of operation and increased power output in modern engines have brought bearing problems to the front. Bearing and oil temperatures found in the high-speed cars of 1935 are in many instances above the safe range for babbitt metal. Because of the low melting point of the tin compounds in babbitt, it has been necessary to develop other bearing materials that have a higher range of working temperature.





Superchargers, designed by Auburn (left) and Graham (below) greatly increase the engine power at high speeds. One problem in the design was to eliminate noise in the rotor step-up drive

Developments in Bearing Metals

Increased speeds, as early as 1929, had demanded bearings that would withstand higher temperatures. A reduction of wall thickness in the babbitt lining from $\frac{1}{16}$ in. to about 0.015 in. increased the resistance to deformation and also improved the flow of heat from the lubricant. Steel-backed, thin wall bearings also came into use about the same time, giving increased stiffness.

All-bronze bearings have sufficient load-carrying capacity at high temperatures and high speeds but they require much harder shafts and their cost of machining is higher. To overcome the necessity of high shaft hardness there was developed, primarily for airplane engines, a copper-lead bearing material in which the copper and lead are mechanically mixed, not alloyed. After several years of development in airplane motors, this bearing material was used last year in automobile engines. With copper-lead linings bonded to steel, this bearing construction combines stiffness, good heat conductivity, and low coefficient of friction with much higher safe working temperatures. Unfortunately, most types of breaking-in oils attack to some degree the free lead in the copper-lead bearings.

Two cadmium bearing alloys, cadmium-silver and cadmium-nickel, have been developed recently, allow-

STUDEBAKER 1929

1935

ing much higher safe working temperatures than can be used with babbitt. These materials cost more than babbitt but less than the copper-lead mixture; like the copper-lead mixture, they are also attacked by some breaking-in oils. However, the cadmium alloys will carry heavy bearing loads at temperatures above the melting point of babbitt, do not require extremely hard journals.

As compared with ordinary babbitt, the cadmium-nickel alloy has the following properties:

	Cd-Ni	Babbitt
Melting Point, deg. F.	600	475
Compressive strength, lb. per sq.in.	26,000	11,000
Ductility, per cent	19	10
Brinell hardness	33	24

Since the cadmium fuses directly to the steel backing without the use of solder or tinning, the bearing will safely withstand temperatures up to the softening point of the bearing alloy. As compared with the cadmium-silver alloy, the cadmium-nickel bearing is said to allow somewhat higher oil temperatures. Both materials, however, are successfully meeting temperature conditions existing in severe service. One company in the commercial field has this year standardized on the cadmium-nickel main and connecting rod bearing for all truck and taxi-cab engines.

For several years, exhaust valves have been made of special tungsten and chromium alloys for increased resistance to oxidation and scaling. For passenger car use, these exhaust valves have continued as standard. Under prolonged service at full loads such as is common today in trucks and buses, the high exhaust valve temperatures may cause a progressive change in physical properties, and a decrease in toughness. Many heavy-duty engines are now using a stainless steel for exhaust valves, although the material costs about three times as much. Since lower working stresses are necessary with the stainless steel, the valves must be designed heavier, and a larger stem diameter is required. It is claimed that this valve can successfully withstand the most severe operating conditions.

Screwed-in ring with hard facing for exhaust valve seats in White truck engines

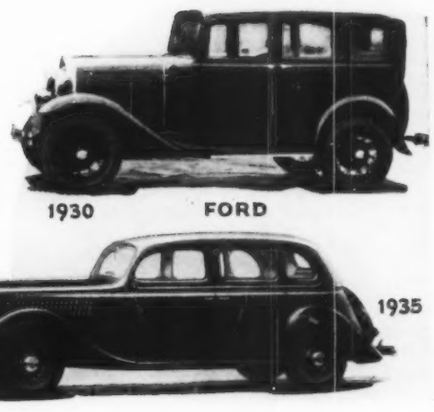
Valve spring surge, first recognized as a problem several years ago, is now being controlled by two methods: friction damping and variable-pitch springs. In several designs a collar slipped over one end of the spring has fingers which rub against some of the coils. Rubbing friction absorbs enough energy to prevent excessive surge or vibration. The most common method used in 1935 cars is to wind the valve spring coil with the turns spaced closer together near each end of the spring. The spacing is arranged so that the end turns close solid when the valve lifts, decreasing the number of active turns, and thus increasing the natural frequency of vibration. Since the vibration period is constantly changing while the valve is being lifted, there is insufficient time for surge to develop.

Use of Valve Inserts

Valve springs plated to prevent fatigue. Inserts for increased wear resistance

Valve spring failures, even when surge is controlled, are sometimes caused by localized stresses set up by quenching cracks, surface defects and impurities in the spring wire. Cadmium plating is used to some extent to prevent formation of rust and possible fatigue failure. For commercial and heavy-duty engines, where spring failure may cause expensive delays, the valve-spring steel is carefully selected for purity, surface finish and ability to harden uniformly.

Rapid wear on the valve seat with its attendant loss of compression and engine performance, was among the first difficulties experienced as a result of the higher speeds. The exhaust valve seats in ordinary cast-iron cylinder blocks of high-speed, high-compression engines wear rapidly under the higher temperatures accompanying continued operation at high speeds. Better valve cooling is one method of improving the valve-seat life. But to improve the valve-seat material, either a better cylinder block material or inserted valve seats of a hard wear- and heat-resisting material were required. By 1932 inserted exhaust valve seats had appeared on a number of passenger cars, and in 1935



they are quite general. These seats are made of tungsten-chromium tool steel and are assembled by a light press fit after the insert has been cooled below room temperature by immersion in solid carbon dioxide. Susbsequent expansion gives a tight press fit.

In 1931 Stellited valve seat inserts first appeared on trucks and are now used on a number of trucks and buses. Records as high as 250,000 mi. for the life of such valve seats have been reported. But the material is expensive and its use on passenger cars, where the expected life is only about 50,000 mi., is not justified, in the opinion of most automobile engineers. On the other hand, for aircraft and diesel engines, where the service is more severe, Stellited valve seat inserts are finding wide acceptance.

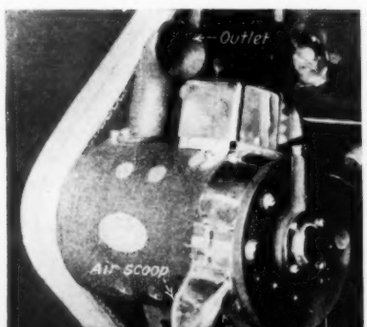
Harder alloys for the block casting also serve to increase the life of valve seats, in addition to greatly diminishing cylinder wear. In combination with directed cooling of the exhaust valve passages and seats, the hard cast iron is giving satisfactory exhaust valve life on a number of passenger car engines.

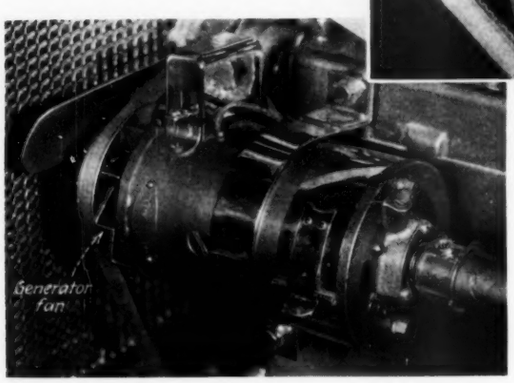
Intake valve seats, for which inserts are not used because they are not subjected to the high temperature, have a greatly increased life in hard cast-iron cylinder blocks. The better distribution of carbon in these alloy cast irons, and the finer grain size likewise prevent the formation of "warts" and the scoring of the exhaust valve seat, while the greater hardness of the material also reduces the tendency to pound out or "mushroom" the intake valve seat, another contributing factor to increased valve seat life.

Superchargers and Generators

Supercharging, another recent development, not only increases the horsepower and gives better performance at higher speeds, but also permits a considerable reduction in motor weight for a given output. In depending on suction for the fuel intake on the suction stroke, the amount of fuel that can be delivered is restricted by the frictional resistance to flow through the carburetor, manifold and valve ports. At higher speeds this becomes more pronounced, decreasing the hp. per cu.in. of piston displacement. By blowing air through the carburetor, the pressure in the manifold can be increased, thus crowding a heavier charge of fuel into the cylinders.

For special purpose designs such as racing cars, supercharging has been in use for a number of years. This year two passenger car makers have adopted supercharging in order to obtain higher horsepower for better acceleration at normal cruising speeds of 50 to 60 m.p.h. and higher. Superchargers give more horsepower for a given engine or make possible a smaller and lighter engine for a given horsepower. With increasing trends toward reduction in weight, greater emphasis will undoubtedly be placed on forced air charging or superchargers.



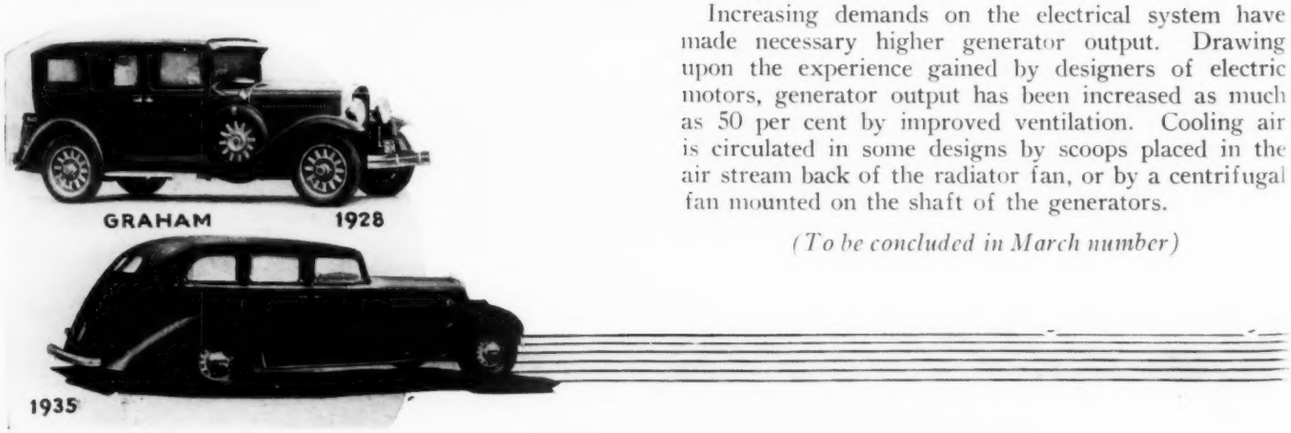


Forced ventilation permits increased generator output —Packard at right, Graham below

Both of the two standard superchargers now offered are of the centrifugal type, running at 5 to 6 times engine speed. Vanes and rotors are of aluminum for reduced inertia and easier balancing, and the blower casing is cored for cooling water. Force feed lubrication is employed, and special attention is given to quiet gear drives. Helical bevel gears and a planetary roller drive are used in the Auburn design, while the Graham impeller is rotated by a single step-up helical gear and pinion drive.

Increasing demands on the electrical system have made necessary higher generator output. Drawing upon the experience gained by designers of electric motors, generator output has been increased as much as 50 per cent by improved ventilation. Cooling air is circulated in some designs by scoops placed in the air stream back of the radiator fan, or by a centrifugal fan mounted on the shaft of the generators.

(To be concluded in March number)



Factors in Redesigning For High-Strength Steel Sheets

When designing for light weight, unit stress is only one of numerous factors to be considered, some of which cannot be calculated

V. H. LAWRENCE
Metallurgist, Alan Wood Steel Company

GREAT interest was created during 1934 in high-strength steel sheets, primarily by the wide publicity given to new railroad equipment that featured light-weight design and high-strength materials, of which the Burlington Zephyr is the best known example. Further interest in this subject was fostered by the development and sale of commercial high-strength steels by three or more producers. Because of the anticipated sweeping effects this new development will have, the writer during the last few months visited a number of engineers, fabricators and manufacturers to ascertain as accurately as possible the future of high-strength sheet steels, the manufacturing limitations of these steels and, primarily, the engineers' viewpoint as to the value of high-strength sheets.

Lack of rigidity may have a bad psychological effect

The following is largely a résumé of the points discussed, arranged in sequence as related to definite engineering factors. It presents the pertinent points the engineers, fabricators and manufacturers, either by experience and hearsay, anticipate or fear through the growing use of high-strength steels. Lest the comments herein which are not wholly favorable or enthusiastic might be construed as unfair criticism of a new product whose manufacture has so far been limited to a relatively few producers of steel, the writer wishes to go on record as being associated with a producer who has for some years, and is now, producing high-strength sheet steels.

In the personal opinion of the writer there is a large field for high-strength sheet steels. As a question of economics, the use of strong light-weight materials is desirable from the standpoint of conservation of raw materials suitable for the production of ferrous metals. In the case of railroad rolling stock and vehicles, the further conservation of fuel justifies taking full advantage of the maximum safe strength possibilities of high-strength sheet steels. On the other hand, ill considered applications leading to whole or partial failure will discourage not only those immediately connected with the failure, but others who learn of the failure without appreciating the reason thereof.

One of the bright spots favoring the more extended applications of high-strength sheets is the fact that many existing applications of sheet steel are unnecessarily heavy, and could be reduced in thickness and weight successfully with standard materials. The attention focused on weight reduction will undoubtedly result in high-strength sheet steels being used on such applications and being credited with the advantage accrued.

Unfortunately, over-enthusiasm usually results in the improper or ill-advised use of a new material. For this reason there can be anticipated a setback in the growth of popularity of high-strength steels. This is expected to be caused by the tendency to base applications on unit stress alone, without regard, or with insufficient regard, to the other features involved. To bring these points to the attention of prospective redesigners, the writer is listing those factors involved.

1. **Establishment of a safe unit fiber stress is affected by:** (*a*) Tension properties—particularly the value of the elastic limit or yield point or proportional limit; (*b*) corrosion resistance and its effect on the unit area (and inversely on fiber stress) after a period of service; (*c*) rigidity—and it is to be noted that no type of steel or method of treatment affects the stiffness. The modulus of elasticity, *M*, remains the same at 29,000,000 to 30,000,000; (*d*) effects of fabrication on static and dynamic properties, such as the favorable or detrimental effect of cold bending, punching or welding on the tension or impact properties of the steel; (*e*) fatigue properties and the relation of live to dead loads.

2. **Determination of those engineering factors not calculable by formulas, including:** (*a*) desirable factors of safety in relation to importance of the member; (*b*) the protection offered during service life by painting, coating and inspection; (*c*) the necessity of rigidity in excess of that required for

strength or deflection as a psychological factor in service; (*d*) the relation of thickness and rigidity to objectionable phonetics. For example, the increased tendency towards echoes and drumming, resulting from a light wall thickness insufficiently supported.

Considering the above items in detail and their interrelation, the relation of safe working load to static tension properties should be considered.

On commercial structural steel, hot-rolled, such as is covered by A.S.T.M. Specifications A7 or A9; Federal Specifications QQS-711 or QQS-721, A.S.M. Class A, etc., the following relations exist between static physical properties:

Yield Point (elastic limit) = ½ Tensile Strength
Elongation in 8 in. (ductility) = 1,500,000 ÷ Tensile Strength.
Safe Working Load = ⅓ Tensile Strength or ½ Yield Strength.

Using A.S.T.M. A7-33 as an example:

Tensile Strength Specified is 55—65,000 lb. per sq.in.
Yield Point Specified is ½ T.S. but 30,000 lb. per sq.in. min.
Elongation Specified is 1,500,000 ÷ T.S. or 23 per cent min.
Safe Working Load (American Institute Steel Construction) is 16,000 lb. per sq.in.

For years, basing experience on steels of the same general characteristics, the safe working load was assumed as approximately ¼ the maximum ultimate strength. Then it was pointed out that the safe working load had no real relation to the ultimate strength, but should be considered more accurately as one-half the yield strength which on the then standard steels, gave the same values of safe working stress. The reasoning is obviously correct. Any load exceeding the yield strength introduces permanent set which constitutes failure of the part or throws the load on adjacent parts. The full realization of strength value of high-strength steels, on which the structural relationship varies from standard steels, demands the acceptance of this metallurgically accurate method of arriving at safe working loads.

Drumming and echoing caused by large unsupported areas is generally objectionable

To show the significance of a proper selection of the safe working stress, the tension properties of the three better known high-strength steels *A*, *B* and *C* are:

	Steel A	Steel B	Steel C
Yield Point	50—60,000	70,000 min.	70—85,000
Tensile Strength	65—75,000	90,000 min.	90—105,000
Elongation in 2 in.	27—22 per cent	20 per cent min.	20—26 per cent

Using the extremely high strength steels *B* and *C* as examples, if the safe working load is assumed as one-quarter the maximum ultimate strength or approximately 26,000 lb. per sq.in., an increased strength of only about 60 per cent or a decreased area of 37 per cent can be obtained.

If, however, a safe working load of one-half the yield strength is assumed, or 35,000 lb. per sq.in. for steels *B* and *C*, a designing advantage of 235 per cent over structural steel which has a yield point of 30,000 lb. per sq.in., is permitted. This use of a 35,000 lb. per sq.in. fiber stress still permits the same factor of safety based on permanent set, as obtained with a stress of 14,000 lb. per sq.in. in A.S.T.M. A7 steel wherein a fiber stress of 16,000 lb. per sq.in. is used in conjunction with a yield strength of 30,000 lb. per sq.in.

As an example, using a section ⅛ in. x 1 in. high-strength steel instead of a ¼-in. x 1-in. structural steel section, a comparison of safe working loads would be:

	Steels B & C ⅛ in. x 1 in.	A.S.T.M. A7 ¼ in. x 1 in.
Safe Working Load	4,375 lb.	4,000 lb.
Failure due to permanent set would occur at	8,750 lb.	7,500 lb.

However, corrosion with a resultant reduction in cross-sectional area is a factor. If the corrosion rate were the same for both of the above steels, then the point at which the cross-sectional area had been reduced to the extent where the full yield strength existed in the structure would be arrived at sooner in the light section of high-strength steel. This follows from the fact that an equal reduction in thickness represents a greater percentage loss for the thinner section.

When increased tension properties permit a decrease in thickness of 40 per cent for example, from ¼ in. to 10 gage or from $\frac{3}{16}$ in. to 12 gage, an increase in resistance to corrosion, or abrasion, of at least 66⅔ per cent is essential to insure the same life in service.

Corrosion resistance can scarcely be predicted, and obviously to hold up a new steel for five to ten years to permit predictions on the basis of service tests is uneconomical. Hence predictions of corrosion resistance are based on the results of accelerated corrosion tests.

Because any chemical, thermal, or mechanical treatment does not change the modulus of elasticity, where the stiffness must be maintained the same moment of inertia in the redesigned section must be obtained through the use of ribs, beads, corrugations, and similar constructions.

Intimately connected with the redesign for rigidity is the ability of the steel under consideration to be fabricated to the new design on existing or available equipment without serious disturbance of the physical properties of the steel by such fabrication. Some steels cold-work harden readily, and the work hardened area may increase in strength with a greater increase in brittleness to the point of susceptibility to service failure through shock or fatigue. Failures caused by stress concentration often fall at areas that are subject to work hardening. Some steels are radically affected by hot working, particularly those steels whose properties are obtained by previous heat-treatment.

Welding as a means of fabrication is so common that its effect on the finished structure must be considered. This must be considered not only where welding is contemplated as a part of the original fabrication procedure, but also with respect to the possibility that welding might be used as correction in case of fabricating mistakes or in service repairs later.

Successful fusion welding of sheet steel involves greater restrictions on the parent metal than are required for acceptable welds on structural members or heavier flat rolled sections. This is partly because the sheet steels are readily adaptable to automatic welding where no allowances can be made for metal defects and the necessity on light material of producing not only a sound strong weld but a smooth appearing weld as well. Also the air hardening possibilities of thin sec-

tions where the air quenching effect is more pronounced and no self-annealing is encountered must be considered.

For fusion welding heavy sections and plates, the following maximum limits in chemical composition are generally adopted:

Carbon	not more than 0.31 per cent
Manganese	not more than 0.75 per cent
Silicon	not more than 0.25 per cent
Sulphur	not more than 0.05 per cent

The relation of welding to corrosion resistance must also be considered. Is a welding rod (if used) available which will result in welds of equal corrosion resistance in the deposited metal, or can a bead of sufficient area and density be produced to permit the use of ordinary rod? Is there a possibility of electrolysis between parent metal and deposited metal caused by differences in potential developed by different chemical compositions? Laboratory tests or accelerated corrosion tests again must be relied upon for answers to these questions.

The most important feature in welded construction—the effect of the weld on adjacent parent metal—requires some consideration of the methods of production of high-strength steels, and the writer will digress for a moment to consider the means by which high unit strengths are obtained.

High unit strength in any of the ferrous material is a function of one or more of the following factors: (*A*) chemistry; (*B*) heat-treatment; (*C*) mechanical treatment.

The advantages and disadvantages of each method of producing high-strength steel are summarized as:

(A) Chemical Composition—The strength (static tension) may be increased above that of pure iron (40,000 lb. per sq.in.) by the addition of hardening elements. Carbon, manganese, silicon, chromium, nickel, molybdenum and vanadium are the most common ones. Carbon will increase the ultimate strength to the extent of about 1,000 lb. per sq.in. per point (0.01 per cent) of carbon and increase the elastic limit (yield point) about 600 lb. per sq.in. per point. Manganese will increase the ultimate at the rate of 100 lb. per sq.in. per point manganese, and increase the elastic limit 60 lb. per sq.in. per point, with the other elements in decreasing value.

The simplest method of producing high-strength sheet is to combine the above hardening elements to produce the desired ultimate strength; and hot-roll and normalize or anneal. To produce material of extremely high elastic limit requires, therefore, an extremely high ultimate strength with correspondingly low ductility. The major objection to this type of steel is brittleness adjacent to a weld, because the high concentration of hardening elements air-quench to extreme brittleness from welding temperatures.

(B) Heat-Treatment—The same values in ultimate strength, higher values in elastic strength and therefore higher safe working loads can be obtained by quenching and tempering steels much lower in alloy or carbon concentration. Better ductility and higher fatigue values are exhibited by the quench and draw treatment. The objections to it are extremely high cost and the inability to obtain a reasonably flat finished sheet.

(C) Mechanical Treatment—The high ratio of elastic limit to ultimate strength on the existing high-strength steels indicate the existence of internal stress, probably set up by some degree of cold work or sub-critical rolling followed by partial stress relieving. Where the relation of cold work to base metal chemistry can be controlled, so that the annealing effect of weld temperatures balances the air-quenching effect inherent in the chemical composition, a high degree of weld strength is to be anticipated.

Of the engineering features not calculated by formulas, two points warrant repetition for emphasis.

The psychological requirement of rigidity often in excess of that required for strength is important. The mental insecurity of the individual who is in contact with a structure of considerable resiliency, even though excessively strong, may lead to a considerable reluctance towards further utilization of such a structure.

Noise, drumming and echoing caused by decreased metal thickness over large unsupported areas may be anticipated in some instances, and is generally objectionable. The cost and weight of silencers may often overbalance the anticipated saving in weight or cost.

The ultimate decision as to the use of light, high-strength steels and the selection of the particular type of steel for the purpose can be more intelligently done by considering which points are important and then verifying them either personally or by reliance on responsible authorities. The following summarizes these features and states where and how they may be verified.

1—BY THE ENGINEER

(a) Can the design be modified to permit substantial saving in weight or cost or additional factors of safety?

(b) Will reduction in metal thickness conflict with any existing insurance or inspection standards, and if so are the advantages to be accrued sufficient to justify an attempt to have these limits modified?

(c) Can the required information from the laboratory, shop and producer be obtained and relied upon?

2—BY THE BUYER'S LABORATORY

(a) Are the physical properties uniform and within reasonable relation to those anticipated or guaranteed?

(b) Does the chemical composition conform to that on which corrosion predictions have been made?

(d) Are shop welds sufficiently dense (free from porosity) to permit comparison with weld corrosion predictions?

(e) Are the physical properties objectionably affected by anticipated methods of fabrication—cold work—hot work—punching or welding?

3—BY THE BUYER'S SHOP

(a) Is the finish available suitable for the part?

(b) Is the flatness available suitable for the part?

(c) Can the material be processed with existing equipment or will new, stronger equipment be required in addition to that which may be required by a redesign using structural steel?

(d) Can the new material be welded with available facilities to produce a smooth workmanlike weld?

4—BY THE PRODUCER

(a) What is the corrosion resistance of the metal as furnished?

(b) Will the corrosion resistance of the structure be changed by anticipated methods of fabrication, service or repair?

5—BY THE BUYER

(a) Is the new material available at a satisfactory price and within reasonable delivery periods?

(b) Are the necessary complements for construction, bolts, rivets, welding rods, etc., available at reasonable price and within reasonable delivery periods?

(c) Will the advantages accrued by the change in design, through saving in net costs, saving in service costs, saving in weight and advertising value offset the cost of redesign, cost of development and production and hazard of service?

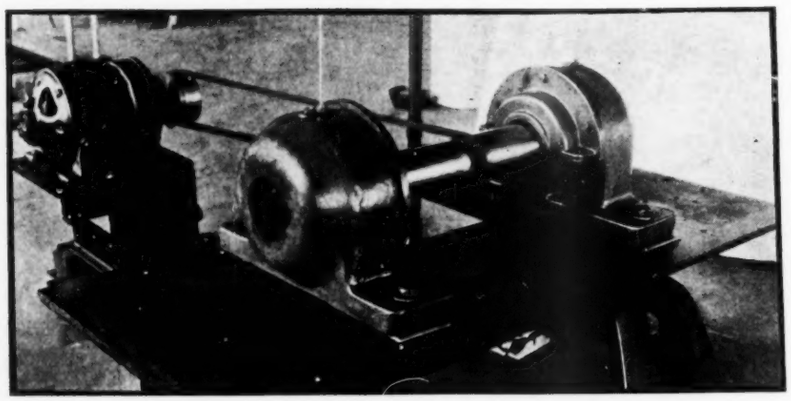


Fig. 1—Testing apparatus for determining effect of type and quantity of grease on the temperature of large double-row bearings operating without load

Part II

Properties Affecting Temperature

Lubrication of Anti-Friction Bearings

Results of tests on grease-lubricated ball bearings, indicating what might also be expected in straight and tapered roller type bearings

F. LEISTER
President, The Bearing Appliance Company

WHEN the machine designer plans to use anti-friction bearings he must give proper consideration not only to the design cost, but also to the service factors involved—operating temperatures, moisture, windage, speed, importance of unit, accessibility, cleanliness, protection from dirt, and fire hazards.

The item of cost is usually an important one. The most inexpensive design is the one using prelubricated bearings with no provisions made for relubricating. At the other extreme is a circulating system with provisions for feeding, regulating and straining the oil automatically. Between these two extremes are many possible variations, depending upon the requirements in relation to the cost of the unit. Simple grease openings are inexpensive and efficient when provided correctly and used properly.

Once it is definitely decided which type of lubrication—oil or grease—is going to be employed for the anti-friction bearings in the new machine, the designer must consider the characteristics and peculiarities of oil or grease in general. Factors that may be objectionable must be controlled and a full appreciation of just what will develop during the operation of the mechanism must be gained, a matter of the utmost importance to the designer.

Effects on Temperature

Most of the variable factors that enter into the problem of lubrication, including the consistency, amount and feed of the lubricant, speed of shaft, and bearing size will have a marked effect on the operating temperature. Within the limits of normal loading, temperature is not affected much by the load itself.

The make-up of a grease, whether it has much or little soap content, and high or low viscosity compounding oil, has its effect on bearing temperature. A large amount of any grease in the bearing housing will induce heat in operation. With a definite amount of grease of a given consistency, the speed at which the bearing runs will also affect the bearing temperature.

Using a predetermined amount of grease, the tem-

TEST NO. 1—Bearing slushed with petrolatum

TEST NO. 2—Bearing lubricated with ½ lb. of fibrous grease, 6 per cent soap, 260 deg. F. melting point, compounding oil 160 sec. Saybolt at 210 deg. F.

TEST NO. 3—Bearing lubricated with ½ lb. non-fibrous grease, 14 per cent soap, 290 deg. F. melting point, compounding oil 300 sec. Saybolt at 100 deg. F.

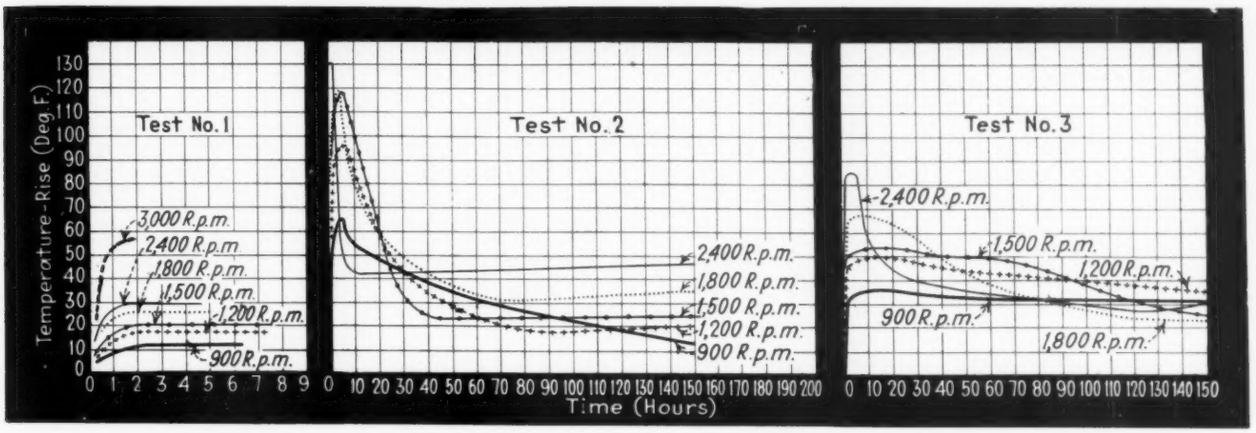


Fig. 2—Results of time-temperature test on No. 5320 double-row bearing

perature at the beginning of operation will be higher than after the machine has been in operation for some hours, and hence the time factor should be given due consideration. Bearing size will also have some effect on the temperature. Rather small or unusually large bearings may require that special attention be given to the lubricating features of the design.

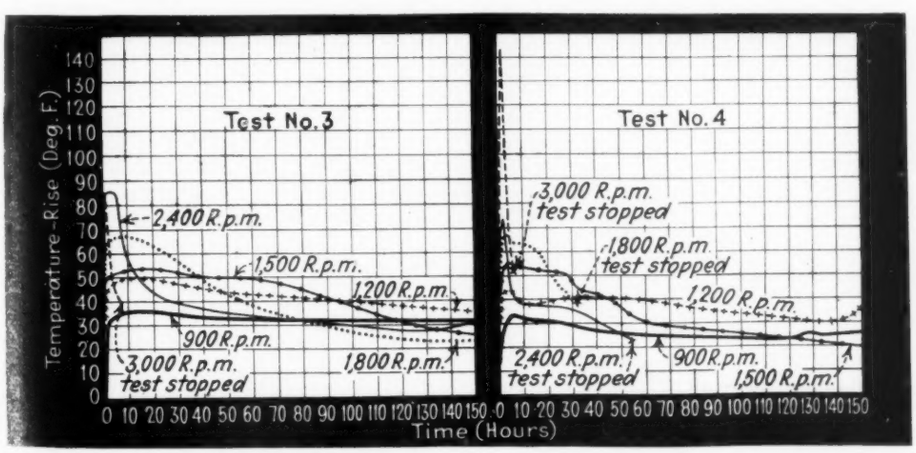
Likewise with oil, the viscosity, amount (oil level and method of application), speed and duration of operation, as well as the size of bearing, will all affect bearing temperatures. As mentioned, load itself, within prescribed and safe limits, has little effect on the temperature.

Actual studies of bearings operating under different lubricating conditions are of particular interest, as documentary evidence of what may be expected. The effect of some of the factors involved is well understood by many bearing users and by all bearing engineers, but some other results are not so generally recognized. A review of these interesting points is therefore presented here.

As a starting point for temperature considerations, it is convenient to make use of the fact that all bearing manufacturers slush their completed product in some mineral protective agent, generally a petrolatum mixture. Since this slush is applied hot, either by dipping the bearing into a heated bath or by spraying it on, the quantity left in the bearing amounts to a light film on all surfaces. Consequently, it may be considered that this is the least amount of lubricant with which the bearing will safely operate, and bearing temperatures resulting from other quantities of grease may be compared with the "slush operating temperature" as a criterion.

Results of Tests

TEST NO. 1—In Fig. 2, Test 1, is shown the temperatures observed in large double-row bearings slushed with petrolatum, when operated at various speeds. The set-up for this test is shown in Fig. 1. It is seen from Fig. 2 that the final rise over room temperature is comparatively slight, but increases as the speed increases, with the bearing attaining its final temperature in two to three hours of operation. It may be further noticed that the higher the speed, the faster is the rise



TEST NO. 3—Bearing lubricated with ½ lb. grease, 14 per cent soap, 290 deg. melting point, compounding oil 300 sec. Saybolt at 100 deg. F.

TEST NO. 4—Bearing lubricated with 20 per cent soap, 300 deg. F. melting point, compounding oil 200 sec. Saybolt at 100 deg. F.

Fig. 3—A comparison of the time-temperature curves obtained from the same bearing when using the same amounts of two greases having widely different melting points and compounded with oils of widely different viscosities

to the final and "leveling off" temperature. Curves of operating temperatures for smaller bearings lubricated only with petrolatum slush will be found on other charts presented here and may be used as a reference from which other comparisons may be made.

TEST NO. 2—With the results shown for Test 1 in Fig. 2 as a basis, the effect of increasing amounts of grease can be ascertained. The results of Test 2 shown in Fig. 2 illustrate an actual comparison. The additional amount of grease need not be excessive, and, for this trial run only one-fifth of a pound was used, since that nearly filled the open spaces in the bearing between the rings, retainers, and balls. With this quantity of grease the same bearings were again operated at various speeds for 150 hr. The resulting temperatures are indicated on the curves.

The relatively great effect of a small addition of lubricant is startling, and may be disquieting to some users. But when it is considered that, as shown by the curves, the temperature falls rapidly because of the expulsion of the lubricant from the immediate proximity of the moving parts, the final effect is seen to be comparable with the slushed condition. Amounts of grease less than one-fifth of a pound could be expected to produce proportional temperatures between the slush and the given condition. For example, it might well be assumed that one-tenth of a pound properly applied to the bearing would give temperature values about midway between those observed in Test 1 and Test 2.

As in Test 1, the results of Test 2 also show that in general, higher speeds cause earlier "peaks" and earlier final temperatures. The long delay in reaching

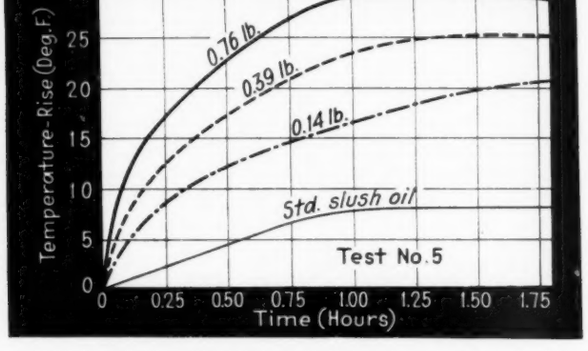


Fig. 4—Temperature curve for a medium-size single row No. 310 bearing lubricated with grease, compared with results obtained when bearing was slushed in petrolatum

the final temperature is strikingly shown for the test run at 900 r.p.m. wherein after 150 hr. a constant temperature had not been reached. On the other hand, for the test run at 2,400 r.p.m., it took only about 12 hr. to reach the normal operating temperature.

The grease used in Test 2 was a fibrous grease with 6 per cent soap (soda base) and a melting point of 260 deg. F. The compounding oil was cylinder stock with a viscosity of 160 sec. Saybolt at 210 deg. F.

TEST NO. 3—Using the identical experimental set-up as pictured in Fig. 1, a different grease was used, and the bearings again operated for 150 hr, at the same speeds as used in Test 2. The grease was used in the same quantity, namely, one-fifth of a pound, but its composition was 14 per cent soda soap, its melting point was 290 deg. F., and the compounding oil had a viscosity of 300 sec. Saybolt at 100 deg. F. The results of these runs are also shown in Fig. 2, and can readily be compared with the results of Test 2.

In Test 3 the high speeds again caused early "peaks," but much lower maximum temperatures than with the grease used in the previous run. The variation noticed in Test 3 during the 150-hr. runs is probably due to working and reworking of the non-fibrous grease.

TEST NO. 4—A still further comparison was made with a third make of grease, the results of which are indicated in Fig. 3, wherein is also shown the results of Test 3 to facilitate making comparisons. The quantity and the application of grease was identical in Tests 3 and 4, as were all other conditions during the 150-hr. runs. The erratic nature of the temperature readings indicates quite some difference in the grease itself. This grease had a soap (soda) content of about 20 per cent, a melting point of about 300 deg. F. and compounding oil of 200 sec. Saybolt at 100 deg. F.

Figs. 2 and 3 indicate how different greases will cause wide variations in the operation of identical bearings, as indicated by rise in temperature over room temperature. All greases used in these tests are considered satisfactory as anti-friction bearing lubricants and are in wide use.

The addition of a small quantity of grease, one-fifth of a pound in these tests, caused some temperature rise. But the tests are more interesting in divulging the differences—and similarities—between various greases and lead to several conclusions, the first of which is:

GENERALIZATION NO. 1—With a great many greases early operating temperature may be high, as is apparent from these test runs, and can be expected frequently.

Effect of Grease Quantity

In Fig. 4 is shown the results of Test 5, wherein, using a given grease and operating at 1,800 r.p.m., the temperature effect of a given quantity of grease in a medium-size single-row bearing, No. 310, was determined. As before, the temperature of a slushed bearing is shown for comparison, which indicated a rise of about 8 deg. over room temperature, while quantities of 0.14, 0.39 and 0.76 lb. cause rises of approximately 21 deg., 25 deg. and 30 deg. respectively, in two hours. This clearly illustrates the effect of over-lubrication, as indicated by the operating temperature, and leads to the second generalization:

GENERALIZATION NO. 2—The greater the quantity of grease used, the higher the temperature reached and the earlier it becomes apparent.

In the March number the author will give the results of five more tests by which he establishes three further generalizations.

Lateral Vibration of Shafts

Simplified equations for the critical speed of lateral vibration of shafts. A second article will give the method for determining the coefficients in these equations

LEWIS F. MOODY
Professor of Hydraulic Engineering, Princeton University
Consulting Engineer, Baldwin-Southwark Corporation

IN THE design of high-speed machinery, the perplexing problems of vibrations and critical speeds can be greatly simplified by considering the elementary forces involved. Of the several kinds of vibrations, lateral, longitudinal and torsional, the first is the type most commonly encountered. Calculations for lateral vibrations and critical speed must be made not only for high-speed machines, but also in the design of slower ones. Any shaft carrying heavy masses will be subjected to lateral vibrations by reason of the flexure of the shaft. Whether or not such vibrations will be severe depends on how closely the speed of rotation approaches the critical speed as determined by the shaft dimensions and the modulus of elasticity of the material of the shaft.

An understanding of vibrations can be simplified by considering the forces that cause the vibratory motion. In Fig. 1 is shown diagrammatically a revolving shaft carrying a wheel weighing W lb., i.e., of mass W/g. As indicated in the figure, the wheel rotates about an axis that coincides with the axis of the bore of its hub, and simultaneously the wheel revolves about the axis as determined by the shaft bearings. The radius of revolution of the center of gravity of the wheel about the axis of the bearings is evidently the deflection y of the shaft, plus the eccentricity e caused by the imperfect balance or mounting of the wheel.

Let F be the centrifugal force of the revolving and rotating wheel. Neglecting the weight of the shaft, the deflection of the shaft will be proportional to the force F causing the deflection, that is:

$$y = KF \text{ or } F = y/K \text{ or } K = \frac{y}{F} \qquad (1)$$

This force F will be equal and opposite to the force exerted by the elasticity of the deflected shaft in its attempt to straighten. The centrifugal force developed by the mass W/g rotating at a radius $(y + e)$ at a speed ω radians per second, is:

$$F = \frac{W}{g}(y + e)\omega^2 \qquad (2)$$

When the system is in force equilibrium y will have a constant value and the centrifugal force exerted by the mass of the rotating wheel will be balanced by the restoring force exerted by the shaft by virtue of its stiffness or resistance to deflection. Hence:

$$y/K = \frac{W}{g}(y + e)\omega^2$$

$$\frac{gy}{KW\omega^2} = y + e$$

$$y = \frac{e}{\frac{g}{KW\omega^2} - 1} \qquad (3)$$

From the above equation it is evident that $y = 0$ when $\omega = 0$; y increases with ω until $(g/kW\omega^2) - 1 = 0$, or $g/kW\omega^2 = 1$. For this last condition, the denominator in the right-hand member of Equation (3), equals zero, and for any finite value of e, however small, y becomes infinite. The value of ω at which this takes place is called the critical speed.

It will always require an infinite length of time for y to become infinite, and it requires an appreciable time for the magnitude of y at the critical speed to grow to an extent that would have destructive effects. Therefore, it is possible to increase the speed of rotation continuously and pass through the critical speed to a higher

speed. But operation at the critical speed must not be permitted to continue long enough to allow the forces and deflections to grow to a dangerous value.

If the critical speed can be safely passed, the value of y again becomes finite, but negative and decreases in absolute value as the speed is raised, approaching $-e$ at very high speeds, when the weight tends to rotate about its center of gravity. Thus the operation will be stable at speeds above the critical.

It was shown above that the critical speed is obtained when the speed of rotation is such that $g/KW\,\omega^2 = 1$. Solving this equation, the critical speed ω_c is found by:

$$\omega_c = \sqrt{\frac{g}{KW}} \qquad (4)$$

Dividing both sides of this equation by 2π gives the critical speed in n_c in revolutions per second,

$$n_c = \frac{1}{2\pi}\sqrt{\frac{g}{KW}} \qquad (5)$$

In order to solve a specific problem, it is first necessary to determine the value of K, which is the ratio of the shaft deflection in inches to the load in pounds required to produce the deflection, as expressed in Equation (1). Of course, the deflection is measured at the point of application of the load, which is also the location of the wheel on the shaft.

According to Equation (1) the factor K is the ratio of the shaft deflection at the section where the force or load is applied to the force causing it. Hence, K is the deflection caused by a unit force. The simplest way to find K is to calculate total static deflection in inches at the disk produced by the weight of the disk, when the shaft is supported horizontally. This method can be applied regardless of the running position of the shaft when in actual operation. For a single wheel, and ignoring the weight of the shaft, the value of K will then be y_s, the static deflection, divided by the weight W of the disk.

Inserting the value $K = y_s/W$ wherein y_s is the static deflection produced by the load W, Equation (5) becomes, for a shaft with one wheel:

$$n_c = \frac{1}{2\pi}\sqrt{\frac{gW}{y_s W}} = \frac{1}{2\pi}\sqrt{\frac{g}{y_s}}$$

In this equation n_c is the critical speed in revolutions per second. The value of g must be expressed in in. per sec. per sec. and is numerically equal to $32.16 \times 12 = 385.9$. To express the speed in revolutions per minute, N_c, we must multiply by 60. Making these substitutions the equation reduces to:

$$N_c = 187.6/\sqrt{y_s} \qquad (6)$$

It is interesting to note that if the revolving weight is in the form of a disk not located midway between the two bearings, any deflection of the shaft will change the plane of the rotating disk. The gyroscopic effects that enter into this tend to resist the change of the plane of the disk, and therefore the change in shaft deflection, thereby raising the critical speed.

All of the above applies to a weightless shaft carrying a single disk. For a uniformly distributed load, such as a heavy shaft without any wheels or disks, and supported by two bearings as a simple beam, the equation for critical speed becomes:

$$N_c = 211/\sqrt{y_{max}} \qquad (7)$$

Numerically, the valve of the deflection y_{max} may be obtained by taking 49.3 per cent, or approximately one-half of the total weight of the shaft, and assuming this weight to be concentrated at the center of the span. The deflection can then be calculated as that of a simple beam with a single concentrated load at the mid point. Generally, for a wheel located centrally on a shaft of uniform diameter the effect of the weight of the shaft can be taken into account by adding half of its weight to that of the wheel and considering the combined weight as if carried by a weightless shaft.

Shaft With Two Loads

In Fig. 3 is shown a shaft in horizontal position and supporting two loads W_1 and W_2. The procedure in developing the equations for calculating the critical speed of such an arrangement is fundamentally the

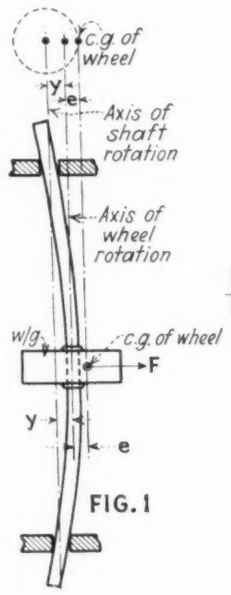


FIG. 1

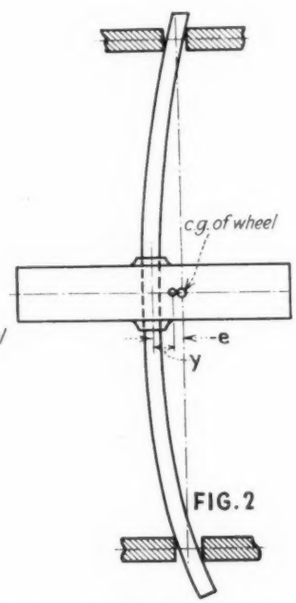


FIG. 2

Fig. 1—The wheel rotates around the axis of its bore and revolves around the axis of the shaft bearings. Fig. 2—At increasingly higher speeds, deflection y approaches eccentricity —e, the wheel then rotating about its center of gravity

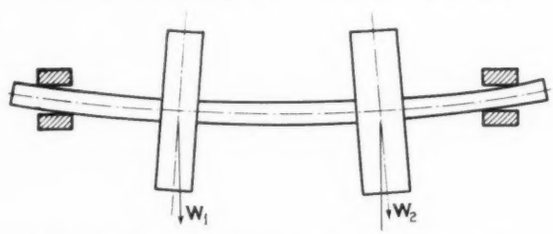


Fig. 3—For a shaft carrying two wheels, the deflection at each load must be taken as the sum of the deflection caused by each of the loads

same as that used in developing the equations for a shaft with a single disk or wheel.

Referring to Fig. 3, the static deflections, caused by wheels W_1 and W_2 located at points (*1*) and (*2*) can be found. Each deflection may be subdivided to separate the effects of the two loads. Thus, a part of the dedeflection at (*1*) is that due to W_1 acting alone, and this is proportional to the load, W_1. Hence this part of the deflection at (*1*) is equal to $k_{11}W_1$; in which k_{11} is the deflection at (*1*) resulting from a load of one pound applied at (*1*). The second part of the deflection at (*1*) is that resulting from the loads W_2 acting alone, and is equal to $k_{12}W_2$; in which k_{12} is the deflection at (*1*) resulting from a load of one pound applied at (*2*). The total deflection at (*1*) will be the sum of the two partial deflections. Similarly, the deflection at (*2*) is equal to $k_{22}W_2 + k_{21}W_1$.

According to Maxwell's "Reciprocal Theorem," or "Principle of Equivalent Deflections," the deflection at (*1*), due to a load at (*2*), is equal to the deflection that would occur at (*2*) if the same load were applied at (*1*). Hence k_{12} is equal to k_{21} and the order of subscripts for the *k's* is immaterial.

Now suppose the shaft is placed with its axis in a vertical position so that no deflections are caused by the weights of the wheels. The shaft is then rotated at an angular velocity ω. Instead of loads W_1 and W_2 acting to deflect the shaft, we shall now have the centrifugal forces of the two masses W_1/g and W_2/g acting to deflect the shaft at the same two points. Calling these forces F_1 and F_2, we can substitute these quantities for W_1 and W_2, respectively. Let y_1 and y_2 be the deflections at the points (*1*) and (*2*). Then:

$$y_1 = k_{11}F_1 + k_{12}F_2 \tag{8}$$

$$y_2 = k_{22}F_2 + k_{12}F_1 \tag{9}$$

The values y_1 and y_2 are the dynamic deflections with the shaft rotating.

Now let us assume that the two masses (*1*) and (*2*) are not perfectly concentric with the shaft, but that one of them, say W_1 is slightly eccentric by the amount e. In the development of these equations, both the disks may be given eccentricities of different amounts, but the final resulting equation will be the same in so far as the critical speed is concerned.

With the given eccentricity, the center of mass of the weight W_1 is now traveling in a circle of radius $(y_1 + e)$ as in the case of the single load previously considered. The centrifugal force developed will be:

$$F_1 = \frac{W_1}{g}(y_1 + e)\,\omega^2$$

Similarly, the centrifugal force of W_2 will be:

$$F_2 = \frac{W_2}{g}(y_2)\,\omega^2$$

Substituting these values of F_1 and F_2 in Equations (8) and (9) we have:

$$y_1 = \frac{\omega^2}{g}[k_{11}W_1(y_1 + e) + k_{12}W_2y_2] \tag{10}$$

$$y_2 = \frac{\omega^2}{g}[k_{22}W_2y_2 + k_{12}W_1(y_1 + e)] \tag{11}$$

From Equation (11) we have:

$$y_2\frac{g}{\omega^2} - k_{22}W_2y_2 = k_{12}W_1(y_1 + e)$$

$$y_2 = \frac{k_{12}W_1(y_1 + e)}{\frac{g}{\omega^2} - k_{22}W_2}$$

Inserting this value of y_2 in Equation (10) and solving the resulting quadratic equation for y_1 by completing the square, we get the final equation:

$$y_1 = \frac{e}{\dfrac{g}{\omega^2\left[k_{11}W_1 + \dfrac{k_{12}^2W_1W_2}{(g/\omega^2) - k_{12}W_2}\right]} - 1} \tag{12}$$

When the denominator of the above Equation (12) becomes zero, the deflection y_1 becomes infinite, even if e is minute in value. This evidently gives the critical or "whirling" speed. Therefore, when the angular rotation ω becomes equal to the critical speed ω_c we have:

$$\frac{g}{\omega_c^2\left[k_{11}W_1 + \dfrac{k_{12}^2W_1W_2}{\frac{g}{\omega_c^2} - k_{22}W_2}\right]} - 1 = 0$$

Solving this equation we get:

$$\left(\frac{g}{\omega_c^2} - k_{11}W_1\right)\left(\frac{g}{\omega_c^2} - k_{22}W_2\right) = k_{12}^2W_1W_2 \tag{13}$$

To simplify calculation let $y = g/\omega_c^2$, giving the equation:

$$(y - k_{11}W_1)\,(y - k_{22}W_2) = k_{12}^2W_1W_2 \tag{14}$$

In this last equation, all quantities except y are constants, and are known from the given conditions. This quadratic equation takes the form:

$$y^2 - (k_{11}W_1 + k_{22}W_2)y + k_{11}k_{22}W_1W_2 = k_{12}^2W_1W_2$$

Solving by completing the square, and resubstituting $y = g/\omega_c^2$ we get:

$$\omega_c = \frac{\sqrt{g}}{\sqrt{X \pm \sqrt{k_{12}^2W_1W_2 + Y^2}}} \tag{15}$$

In the above equation

$$X = \tfrac{1}{2}\,(k_{11}W_1 + k_{22}W_2)$$
$$Y = \tfrac{1}{2}\,(k_{11}W_1 - k_{22}W_2)$$

In the above equation the value of $(k_{12}^2W_1W_2 + Y^2)$ is evidently always positive, giving at least one real value for ω_c, and usually two, indicating that there is a lower and a higher critical speed. In the case of a continuous load on the shaft there are an infinite number of critical speeds corresponding to various harmonics of the lowest or fundamental frequency. In most design problems it is possible to keep the operating speed well below the lowest critical speed, in which

case the higher critical speeds do not enter into the problem. Therefore, for the usual problems, we need only obtain the lowest value of ω_c which corresponds to that obtained from Equation (15) with the plus sign in the denominator.

To express the critical speeds in revolution per minute, it is merely necessary to multiply Equation

Fig. 5—With two bearings between the two loads, both loads must be assumed as acting downward

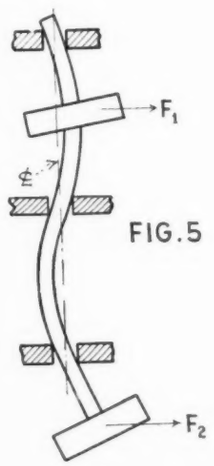


FIG.5

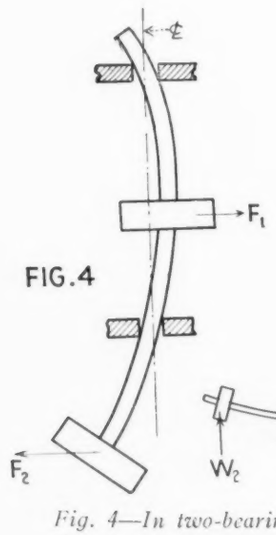


FIG.4

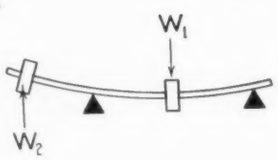


Fig. 4—In two-bearing arrangements such as this the upper and lower forces F_1 and F_2 must be considered as acting in opposite directions so as to result in maximum deflections

(15) by 60 and divide by 2π. If this be done and the numerical value of g in in. per sec. per sec. be substituted, Equation (15) becomes:

$$N_c = \frac{187.6}{\sqrt{X + \sqrt{(k_{12}^2 W_1 W_2 + Y^2)}}} \qquad (16)$$

Let:

$k_{11}W_1 = y_{11}$, the partial deflection at (1) caused by W_1 alone
$k_{22}W_2 = y_{22}$, " " " " (2) " " W_2 "
$k_{12}W_2 = y_{12}$, " " " " (1) " " W_2 "
$k_{21}W_1 = y_{21}$, " " " " (2) " " W_1 "
$k_{21} = k_{12}$

Substituting the above relationships in Equation (16) we get:

$$N_c = \frac{187.6}{\sqrt{\frac{1}{2}(y_{11} + y_{22}) + \sqrt{y_{12}y_{21} + \left(\frac{y_{11} - Y_{22}}{2}\right)^2}}} \qquad (17)$$

In the above problem the two loads were taken within a single span between two bearings. The reasoning is general, however, and is equally applicable to cases where the loads are in different spans or overhung, or where there are three or more bearings. However, care must be taken in determining the direction of the static deflections. That is, the direction of either load W_1 or W_2 must be reversed when, as in an overhung shaft, the one load causes a deflection in a direction opposite to that produced by the other.

It is to be kept in mind that the static deflections are obtained only to determine the relative influence of forces applied at these points, and that these static conditions no longer exist when the shaft is rotating. With the shaft vertical, the centrifugal forces can act equally well to the right or to the left, and in horizontal rotating shafts the centrifugal forces can act either upward or downward. That direction of deflection which will result in the largest deflections will determine the lowest critical speed.

As an example of the above, in the two-bearing arrangement shown in Fig. 4, which represents the shaft of a vertical waterwheel and generator, the force F_2 must act to the left in order to increase the deflection at the upper load which acts toward the right. Therefore, in finding the deflections per unit load, k, the shaft must be considered to be loaded as shown in the diagrams in the auxiliary view.

For a three-bearing unit with rotating masses located as shown in Fig. 5, F_1 and F_2 must be taken in the same direction—that is, both W_1 and W_2 must be applied downward in the static analysis.

Another case is that of two spans with simple loading, shown in Fig. 6. Here the loads act oppositely, for the lowest critical speed. The condition where they act in the same direction corresponds to the higher critical speed.

If the loads are so widely separated with many bearings between that the deflection of one has no appreciable effect on the other, as in Fig. 7, $k_{12} = 0$ and $y_{12} = 0$ and $y_{21} = 0$; which gives:

$$N_c = \frac{187.6}{\sqrt{\frac{1}{2}(y_{11} + y_{22}) \pm \frac{1}{2}(y_{11} - y_{22})}} \qquad (18)$$

$$= \frac{187.6}{\sqrt{y_{11}}} \text{ or } \frac{187.6}{\sqrt{y_{22}}}$$

The above equation shows that the two critical speeds are those caused by each load acting alone. The lowest of the two speeds is then the lowest critical speed of the shaft, unaffected by the presence of the other load.

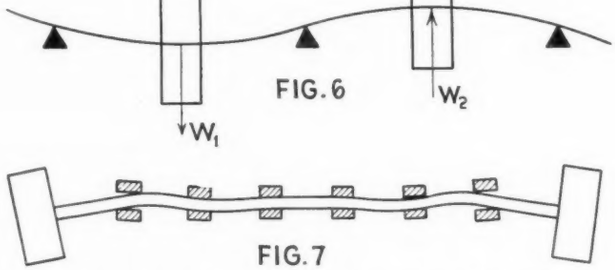


FIG.6

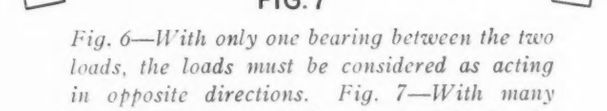


FIG.7

Fig. 6—With only one bearing between the two loads, the loads must be considered as acting in opposite directions. Fig. 7—With many bearings between widely separated loads, the deflection at one load has no appreciable effect upon deflections at the other load

Lighting Requirements

Data on density, diffusion and distribution of light required for local lighting on machines and instruments are given in this second of a series of three articles on built-in illumination

ILLUMINATION requirements for built-in lighting are now well established by scientific tests, as explained in the article, "Design for Better Seeing," in the January number. Accurate data are available on the intensity, diffusion and distribution required for minimum eye strain and maximum accuracy for any lighting installation. The required intensity of illumination for best vision varies greatly according to the type of work being done, the reflectivity and color of the lighted surfaces and the length of time that the eyes are to be focused on the work.

Courtesy Westinghouse Lamp Company

Light intensities of more than 50 foot-candles are needed to give accurate and safe vision needed over the working area of machines

Intensity is measured in foot-candle units, a foot-candle being the illumination intensity given by a one-candle power source at a distance of one foot. In general, intensity is inversely proportional to the square of the distance from the source, and directly proportional to the watt rating of the electric lamp used, since the candle power varies approximately as the rated watts. Diffusion, or the elimination of shadows and intense reflections from bright surfaces, depends on the filament size, type of glass bulb and reflector used, as well as the character of the lighted surface. Reflectors also control the distribution, or the variation in intensity over the lighted area.

For distinguishing large masses or outlines, for lighting up aisles or showing the position of machines, an intensity of 2 to 5 foot-candles supplied by overhead reflectors usually will be sufficient. But when the eye is focused on a smaller field, such as a center punch mark under a drill point, or a graduation on a micrometer or scale, or when surface finish is to be observed, much higher light intensities and better diffusion are necessary.

In the observation of small details, the human eye requires a large increase in light intensity to give a relatively small increase in visual accuracy and a small decrease in eye strain. For example, increasing the intensity from 5 foot-candles to a level of 10 foot-candles doubles the visual accuracy. To again double the accuracy of vision requires 50 foot-candles, an increase of 40 foot-candles. Thus, as more accurate vision, optimum comfort and higher intensities are required, the cost of satisfactory illumination may become excessive if it is supplied entirely from overhead or general room lighting. By bringing the light source closer to the work, smaller lamps of lower rating may be used, reducing the required capacity of the light source inversely in proportion to the square of the distance. In addition to the reduction in size and cost of the equipment, the lower intensity at the source reduces glaring reflections from the surface of objects near the lamp.

For machine tools, such as automatic screw machines, lathes, milling machines and grinders, accurate vision requires a minimum intensity of about 50 foot-candles over the working area, according to recognized authorities in this field. As explained by S. G. Hibben, director of applied lighting of the Westinghouse Lamp Company, the 5 to 10 foot-candle intensity under ordinary conditions in existing installations is totally inade-

quate if real eye comfort and health is a consideration. Jig borers, die-making machines and sensitive drilling machines for small drills will need the maximum intensity listed in Industrial Lighting Codes, or even more—up to at least 150 foot-candles.

Intensity Requirements

Light intensities must be adjusted according to the reflectivity, motion and size of the work. For ordinary metal surfaces, 10 to 60 per cent of the light from a diffused source is reflected. Other materials, such as molded composition, wood or textiles, absorb more light so that a higher level of intensity is required to give the same eye response. The amount of light that actually reaches the eye also depends on the size of the reflector and its shape in relation to the lamp used.

Since the reflectivity of metal surfaces like steel does not vary greatly, the proper level of light intensity for machine tool applications can be provided in the design by mounting a lamp of the required watt rating at a fixed distance from the work. For the intensity range needed for accurate work the actual light intensity is not critical as long as it is within the required range of 50 to 100 foot-candles, since there will be no great difference in eye response for variations within this range. The distance of the lamp from the working area thus can be varied considerably without causing an appreciable change in vision, so that for convenience the lamps can be mounted on adjustable supports or flexible connections.

Visual accuracy, even with light of the proper intensity, is greatly influenced by glare or eye fatigue. The human eye cannot adjust itself for both high and low intensities at the same time. Repeated changes in intensity also cause fatigue in the adjusting mechanism in the eye, slowing up the natural correction and reducing clarity of vision.

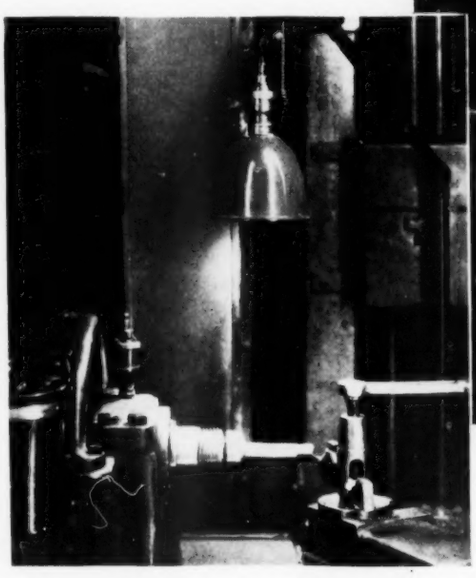
Intensity contrast, the size of the lighted area, light intensity and the length of time influence the amount of glare and the resulting eye fatigue. The contrast ratio between the light intensity in the illuminated area and the background intensity should not exceed 20 to 1 for built-in machine lighting. For a working illumination of 100 foot-candles over a restricted area the general room lighting should exceed 5 foot-candles; with higher intensities the general lighting must be stronger. Gray or white paint on machines, walls and ceilings helps to reduce the contrast ratio, increasing the general level of illumination because of high reflectivity of such surfaces.

Glare also results when high intensity light in the field of vision strikes the eye. Lamp filaments, direct reflections from bright surfaces, sunlight, unshaded windows in the field of vision, in addition to causing eye fatigue will also cause the eye to adjust for a higher intensity level resulting in less visual accuracy over the illuminated working area. Duration or time of exposure intensifies the effect of glare, depending on the nature of the work. If the operator is confined to the machine and must give constant attention to the work, good diffusion and low contrast ratio is as necessary as the proper intensity of illumination.

Light Distribution

With the proper intensity of well diffused light efficient illumination requires only that the light be properly distributed. The working area of the machine should be uniformly lighted, but for economy and protection the light must be confined, using a lamp and reflector combination that gives the desired intensity for the minimum size of lamp.

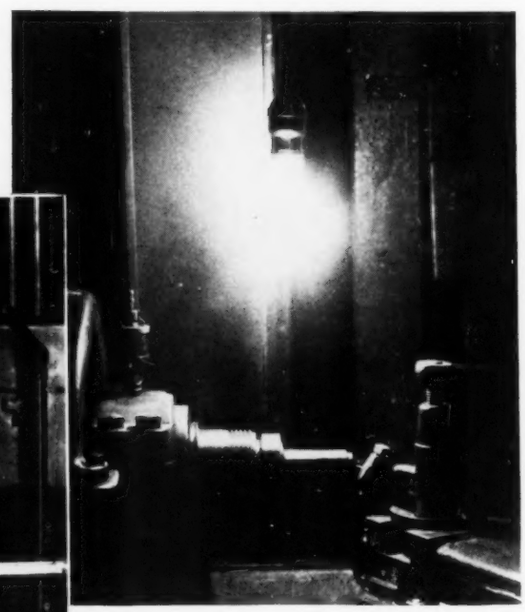
Machine tools represent but one of the fields where local lighting can increase performance and saleability. Accurate seeing is also needed in reading scales and charts on electrical instruments, temperature indicators



Although the lamp shown at the right supplies the required light intensity on the work, extreme eye strain and fatigue will result from the glaring, unprotected light source

•

When directed and diffused by the reflector, the same lamp gives glareless, shadowless illumination, greatly increasing the accuracy and safety of vision



Courtesy Westinghouse Lamp Company

and recorders, weighing devices and many other indicating mechanisms.

For these applications two methods are available for increasing readability; built-in illumination and magnification. Magnification, of course, also requires built-in illumination, and in addition involves such factors as the size of field, space available for the lens, focal distance, and the distance from the lens to the observer. The illumination considerations, however, are the same for a magnified scale as for one that is read directly.

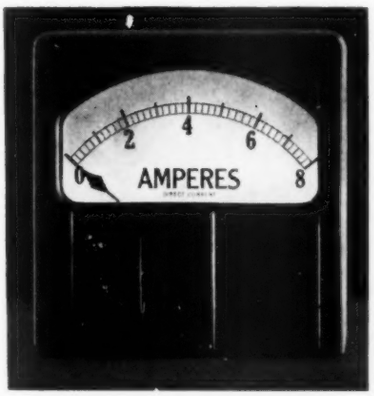
Since nearly all instruments are read through a glass cover or window, reflections from the glass often cause trouble when overhead lighting is used. These high intensity reflections result in glare and decreased visibility of the scale or chart. The designer can supply glareless light of the proper intensity and distribution at much less cost to the user than makeshift overhead lighting; and with proper built-in illumination, the performance of the instrument will not be penalized by poor room lighting or improvised light installations.

Compactness, limited rate of heat dissipation and the high reflectivity of the lighted surfaces in instruments require low-wattage lamps with small bulbs and placed closer to the lighted area. However, the short distance from the lamp to the reflecting surface makes it difficult to obtain a uniform distribution of light.

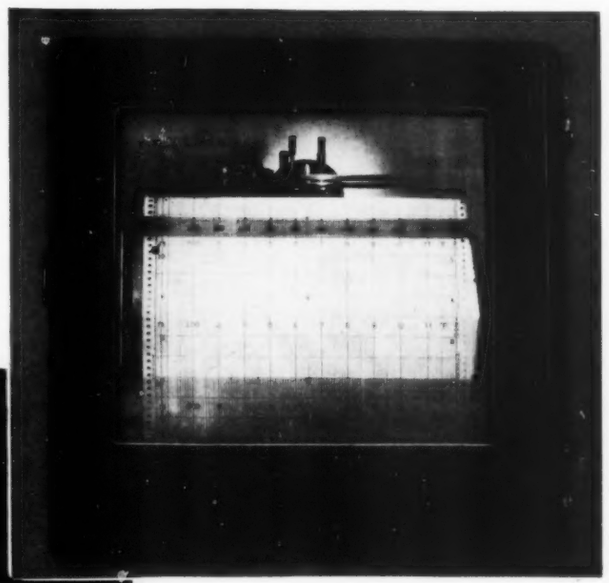
Instrument Applications

Three typical instrument applications of built-in lighting as standard equipment are shown on this page. In the weighing device, light from a lamp in the hood over the instrument is totally reflected, throwing a shadowless diffused light over the entire dial. With the light directly above the dial, no confusing reflections are thrown from the glass window.

Light over the large area of the chart in the recording pyrometer is supplied by a small lamp mounted back of the door frame inside the instrument. Because of the high reflectivity of the paper chart, only a small lamp is needed, while the distance from the lamp to the paper is large enough to give good light distribution. In the ammeter, the light source is of necessity so close to the scale that two lamps are needed to obtain uniform distribution of light. By painting the inside of the case white, the light is diffused so that no shadows are made by the pointer. The high reflectivity of the white enameled scale also reduces the lamp wattage needed thereby reducing the heat generated within the case and temperature effects in the instrument.



Two small built-in lamps illuminate without glare or shadows the scale of the Westinghouse indicating instrument



Mounted behind the upper part of the door frame in the Foxboro pyrometer is a 6-watt, 10-candle-power lamp giving a spread of light that uniformly illuminates the chart and scale without reflections from outside lights

High intensity light without shadows is thrown from reflector mounted above the Roller Smith precision balance

Standard lamps, sockets, switches, wiring devices and lamp supports available and designed for such built-in applications as those illustrated here and in the previous article will be described in the third and final article to appear in the March number. Actual designs, mounting details, and electrical equipment used will be shown for a number of successful applications.

ROBERT G. N. EVANS
Chief Engineer
Bunting Brass & Bronze Company

Information to Give When Ordering Bearings—II

In the January number the author presented a proposed form for use when making inquiries for sleeve-type bearings, and discussed the first seven of the twelve items of the form. The remaining items are explained here

OF ALL the factors that enter into the operation of sleeve-type bearings, none is as important as that of lubrication. There are two phases to this part of the problem. One concerns the type and properties of the lubricant used, the other has to do with the mechanical provisions for applying and distributing the lubricant. The latter will be greatly affected by the type of lubricant used, whether oil or grease.

As to the particular kind of oil or grease that is to be used, that is a question that cannot be considered until all the factors that determine operating conditions and unit bearing pressures have been established. It is then necessary to specify a particular grease or oil, as the case may be, that will have the required characteristics and properties.

8—Character of Lubrication

Factors involving lubrication may be divided into four groups: (*A*) Constant, (*B*) Interrupted, (*C*) Kind of lubricant, (*D*) Grooving.

(*A*) By constant lubrication is meant that the supply of lubricant is maintained at the rubbing surface by a reservoir, automatic oilers or a pressure feed system.

(*B*) By interrupted lubrication is meant that the lubricant supply is limited and depends on the human elements for replenishing. With this type of oiling system it is possible for bearings to go unlubricated for an indefinite length of time. This is almost certain to happen if the bearing is in a remote place.

(*C*) The kind or type of lubricant may be divided into two classes: (1) fluid—oil, light, medium or heavy; (2) non-fluid—grease.

(*D*) Grooving design determines a good or bad bearing if all other factors of the bearing are correct. Considering the duties of grooves, they are needed either to direct or restrict the flow of the lubricant. Oil will flow unaided one-half inch to each side of the point of oil entry along the bearing axis regardless of diameter. If the bearing is longer than one inch plus the diameter of the oil entry hole, it is obvious that grooves are needed. Grooves should not be parallel to the bearing axis unless they are in the region of low film pressure, as previously indicated. Grooves as in Fig. 4 should go out from the oil entry in the direction of travel—capillary attraction will do the rest.

If the direction of rotation reverses, an oil groove is recommended. A loop groove, as shown in Fig. 5, on an angle from the bearing axis, but not allowing the groove to come out the ends, should be used.

The sides of grooves should never be sharp as this makes them act as oil scrapers. Edges should be chamfered if the lubricant is to be distributed over a wide area, and thus allow the lubricant to flow out over the edges and across the rubbing surface. This is also true of the hole drilled for the oil entry. Many bearings are over-grooved, thereby breaking up the oil film and reducing the load-carrying area.

The principle of grooving does not change for different types of lubricating systems. The point of oil entry, however, does. For gravity systems the oil entry

must be on the low pressure side. But for pressure systems the oil entry should be located adjacent to the high pressure area on the inbound side to carry the lubricant into the highest pressure area instead of away from it.

When grease is used the depth of the groove should not equal the width as it is the surface of the grease that furnishes the lubrication. Groove edges should be beveled or chamfered, an extremely important item when grease is used.

When wick oiling is used the grooves should be shallow, as drops, not gallons, of oil are being supplied.

When the bearings are split the edges should be chamfered, but not the entire length. This allows a wedge of the lubricant to form, facilitating surface coverage.

9—Operating Conditions

Surrounding conditions may have a great effect on bearing life, affecting the housing design. This condition can be divided into four parts, as follows: (*A*) Heat; (*B*) Dirt; (*C*) Possible damage to product by oil leaks; (*D*) Acids.

(*A*) Heat. When the bearing is operated in the presence of heat the material must be one that does not lose its physical characteristics at the operating temperature and a special lubricant should be recommended by the oil company.

(*B*) When dirt, grit or any foreign elements are present, a bearing closure must be used to prevent these elements from coming into contact with the rubbing surface or entering the oil supply. This is a factor too often ignored by the machine designer. Proper application of seals will often multiply the life of the bearings and improve machine operation.

(*C*) When the product that is to be fabricated may be damaged by leaking oil, the bearings must be grooved for restricting the oil travel and in some cases a closure designed to retain the lubricant in the bearing mounting. In some cases these closure seals must do both—retain the lubricant and keep out foreign matter.

(*D*) When operated in the presence of acid the metal composition of bearing and shaft must be one that is unaffected by the acids, or corrosion and failure are the results.

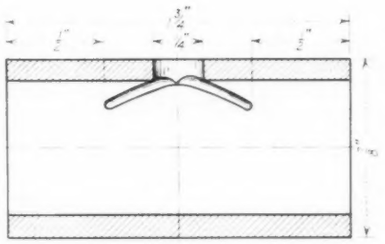


Fig. 4—The grooves should not come closer than ½ in. from the ends of the bearing and should extend circumferentially only in direction of rotation

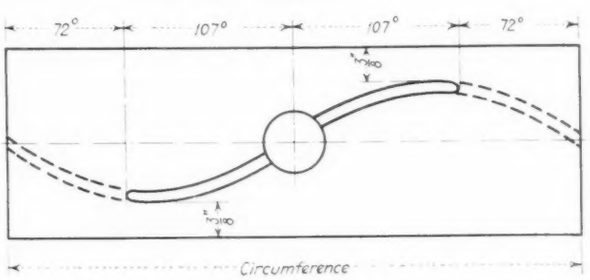


Fig. 5—If grooves were continued as indicated by the dotted lines, they would form a continuous spiral in the bearing

10-11—Space Available

Projected area is length times width and does not take grooving into consideration. For example, for a given design 1 sq.in. of projected area may carry the desired load with a factor of safety of 3 but because of the type of lubricating system the oil inlet must come on the high pressure side and grooves are necessary. With cross grooves ⅛ in. wide, subtracting the area lost by these grooves, there remains 0.8038 sq.in. approximately, and the factor of safety is reduced to about 2.66. The projected area should not be confused by pressure area because due to the different radii of the bearing and the shaft only 40 per cent or less of the projected area will be the pressure area.

12—Unusual Service Conditions

Wall thickness is a dimension that is not given enough thought, as a bearing with all other factors correct can be ruined by an excessively thick wall. The resistance to deformation under load increases as the thickness of the metal decreases, as indicated in Table III.

Table III—Relation of Resistance to Deformation to Wall Thickness

	Deformation Under Loads		
	13,000	15,900	31,500
Specimen 2 in. high, 1 in. diam.	0.001	0.010	0.160
Specimen 1 in. high, 1 in. diam.	0.0019	0.004	0.07
Specimen 1/2 in. high, 1 in. diam.	0.00021	0.00078	0.0084
Specimen 1/4 in. high, 1 in. diam.	0.000	0.0003	0.001
Specimen 1/8 in. high, 1 in. diam.	0.000	0.00005	0.0005
Specimen 1/16 in. high, 1 in. diam.	0.000	0.000	0.000

In designing a bearing the wall thickness should be as heavy as possible for manufacturing and assembling purposes.

While there may be a difference in wear resistance of one gram loss (dry) between one bronze and another, this is not considered in calculating a bearing because lubrication and other factors influence wear rate to such an extent that one cannot be guided by low wear rate comparison.

After considering all the preceding paragraphs it is now possible to determine the composition of the bearing bronze that will give the most satisfactory service. Each of the different factors that must be known before deciding upon the alloy to be used have been explained. With such information the bearing manufacture is enabled to make his recommendations intelligently.

QUESTION AND COMMENT

Tomorrow's Electro-Machine Tool

R. H. CLOBES
Industrial Division
Westinghouse Electric & Manufacturing Company

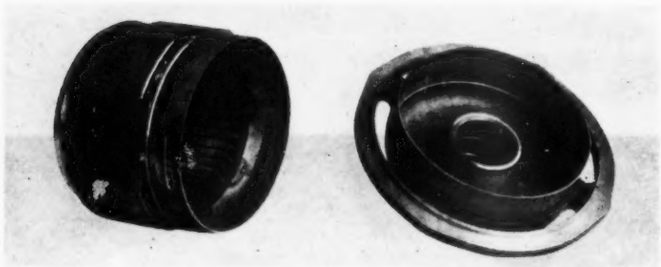


Fig. 1—Prewound stator inserted in a cored hole in the machine frame is inclosed by the copper diaphragm in the end bell

● Nearly every operation in future machine tools will be performed by an individual frameless motor in which the motor shaft is an extension of the machine driving member. Controls and wiring for this drive will be entirely concealed within the machine housing.

For main drives the stator of the motor, as shown in Fig. 1, will have a smooth turned finish over its laminations with the diameter held to a tolerance of a few thousandths. By means of a loose sliding fit, to facilitate removal for repairs, the stator will be held in a cored hole on lands in the machine housing. Two sets screws, 90 degrees apart, will prevent endwise movement and rotation about its axis. The rotor shown in Fig. 2 will be pressed on the spindle or driving shaft and held in place by a key. Balancing rings placed on the rotor will facilitate dynamic balance.

Rotor and stator windings will be entirely inclosed and protected against metallic dust. End rings welded on the stator iron flush with the coil extensions will have a machined section dove-tailing into a corresponding section in a copper diaphragm, making a dust-tight joint. The shaft will pass through a hole in the diaphragm, with a grease or felt seal around the shaft.

Cast projections of the rotor end rings will keep the inclosed air in motion, driving the generated heat to the copper diaphragm. A fan mounted on the front end of the shaft will force air over the outside of this diaphragm, over the stator laminations between the lands, and out through the rear end shield. For auxiliary drives the completely inclosed shell will be used without forced ventilation, the heat being dissipated by conduction through the machine housing.

Controls on this machine tool will be either fully automatic or semi-automatic, operated by pressing pushbuttons. Where there is little depth in the housing, contactors and relays will be mounted on a panel recessed in the casting, with ample room provided within the casting for wiring, inspection, and repair. Where breadth is a limiting factor, but there is ample depth, the contactors and relays will be mounted back-to-back on a panel supported on a sliding rod.

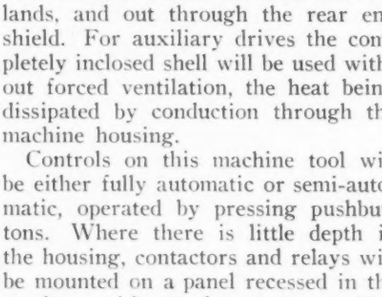


Fig. 2—One-piece rotor pressed on the driving shaft has integral fans for circulating cooling air within the inclosed motor

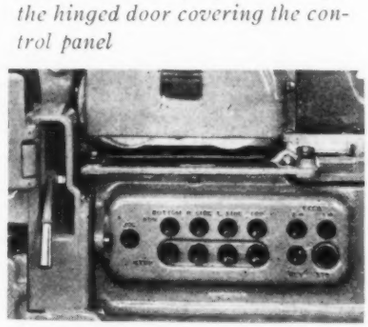
Electrically and mechanically interlocked controls will perform predetermined sequences of operation. On a grinder, for example, the start button will start the spindle motor, the coolant motor, and the table feed motor. The feed motor will feed slowly, reverse, and rapid traverse back to the starting point, and either stop or repeat the cycle, depending on the setting of a selector switch.

Pushbutton stations within convenient reach of the operator will project through holes in a cast cover inclosing the station units, as shown in Fig. 3. Standardized colors on pushbuttons and their wiring will designate the functions. Lead-covered cables will be used for control wiring within the housing of the machine tool to prevent mechanical injury. No flexible or rigid conduit will be used.

Precision limit switches and photoelectric devices will be extensively used, multi-speed prewound stator type motors will replace change gears; thermostatic devices on the laminations or the windings of the motor will protect motor windings, indicating to the operator when the machine is overloaded, so that he can ease up a bit to avoid shutting down the machine.

In addition to improved appearance, the new machine tools will reduce hazards to the operator, making the electrical equipment accessible for inspection and repair, and prevent mechanical injury by concealing the controls and wiring. In the finished product, the appearance of the motor and control will show that it is designed for the particular machine and not applied as an afterthought.

Fig. 3—Pushbuttons for the automatic controls project through the hinged door covering the control panel



Point or Line Contact in Worm Drives?

To the Editor:

● What is the form of contact between the thread of a worm and the tooth of the gear with which it meshes? Is a line contact possible between two convex warped surfaces? Teeth of spiral or helical gears have only a point contact and a multiple thread worm and gear combinations are closely related to helical gears. At what point in the transition from one to the other does it become a pair of helical gears?

In the conventional method of producing a worm and gear, the worm is finished with a lathe tool having a straight-line cutting edge, the tool being pointed toward the center of the work. Thus, on the working surface of the worm or gear, every transverse element is a straight line pointing toward the center line of the worm. The gear hob is chased in the same way as the worm except for increased depth of threads to provide clearance. Because of this construction it is argued that the lines of the gear tooth must lie in the lines of the worm thread.

The point of contact between a worm and its gear gives a better working surface than that between helical gears, because the curvature is less. A quadruple-thread worm has a working surface of much less curvature than that of the single thread, so that the most efficient worms are those having a lead angle of 40 deg. or more.

But a line contact occurs between a worm and gear only when the worm thread becomes parallel with the center line and then it will be something other than a worm. —GEORGE W. HART

Cincinnati, Ohio

Distortion of Hollow Shafts and Pins

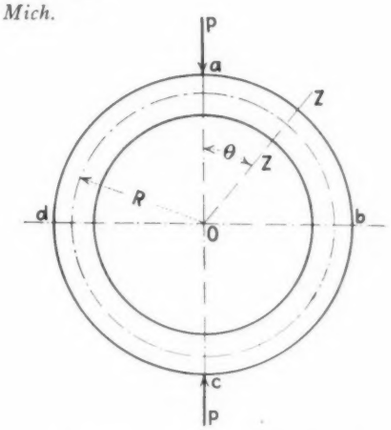
JOHN S. CARPENTER

Detroit, Mich.

● Hollow shafts and pins are finding increased application in modern high-speed, light-weight designs. When one considers that a shaft with a hole half its outside diameter, suffers only a 6 per cent increase in stress with a weight reduction of 25 per cent, the possibilities are worth further investigation.

In addition to any torsional stresses that may be present, there are bending stresses due to transmitting the applied radial loads to bearing shells, rod ends, clevises, and other pin type bearings. Except where there are drive fits, all shafts and pins which transmit radial loads may be considered to have a concentrated diametral load at the point of bearing. The clearance around such shaft or pin, while radially alike over the semi-diameter, increases from nothing at the point of bearing to infinity at the semi-diameter, when considered in a diametral plane coincident with the loading.

Shaft distortion can be calculated by the following relations, based on the Castigliano theorem for statically indeterminate conditions. With a load applied as shown in the accompanying diagram, there exists at any radial section *Z-Z* direct compression, bending and shear, but the shear is small compared to the bending stresses. From symmetry, the compression at the four sections *a*, *b*, *c* and *d* will pass through zero after



Equations for the change in diameter of hollow shafts subjected to bending and shear are given in terms of "R," the mean radius

the distortion, and the bending moment will pass through zero at some point *Z-Z*, the total amounts being equal at *a* and *b*. If M is the bending moment at *Z-Z* at angle θ to the vertical, and M_1 is the bending moment at *b*, the shear being zero, by equilibrium,

$$M = M_1 + \frac{P}{2} R (1 - \sin \theta) \qquad (1)$$

The total bending between *a* and *b* being zero, the bending per unit length (as in a straight beam, *di* being an element of angle, and *ds* an element of length equal to $R\ d\theta$), $di/ds = M/EI$, and

$$\int_0^{\pi/2} \frac{M}{EI} ds = \frac{R}{EI} \int_0^{\pi/2} M d\theta = 0$$

Substituting M from Equation (1) and dividing through by R/EI, this relation becomes,

$$M_1 = \frac{PR}{\pi} \left(1 - \frac{\pi}{2} \right) \qquad (2)$$

Substituting this value for M_1 in Equation (1),

$$M = PR \left(\frac{1}{\pi} - \frac{1}{2} \sin \theta \right) \qquad (3)$$

At the point of load application where θ is zero, M_0 is 0.318 PR. The bending moment is zero where sin θ is $2/\pi$ or 93.5 deg.

Distortions due to bending can be computed using the Castigliano theorem showing that the decrease d_v in vertical diameter and the increase d_h in horizontal diameter is

$$d_v = 0.149\ PR^3/EI$$

$$d_h = 0.137\ PR^3/EI$$

With the distortions in diameter allowable in practice for hollow shafts and pins, especially with regard to seizing or binding in a bearing, the stresses in themselves are rarely important. Only when the wall thickness is great compared to the mean radius do the shearing and longitudinal forces have to be taken into account.

Elastic Modulus of Cast Iron

To the Editor:

● In his article on the elastic modulus of cast iron in the December number, J. M. Lessells is correct in his calculations and his reasoning, but it is not apparent that there is need for very accurate determinations of the modulus of elasticity, especially in tension. The transverse modulus is obtained so easily from the bending curve of the ordinary standard bar that it seems unnecessary to use any other method of determination.

The set at 25 per cent of the load is rather a small proportion of the total deflection, and as most designers would hardly use cast iron at loads higher than one-fourth of its strength, it is hardly worth while to carry out actual set tests. If the amount of set must be known, the tests are easily carried out on the transverse tests by loading the bar a given amount five or six times, and then taking a stress-strain curve up to this point. It would

be sufficient to measure the deflection at the final point and use it as the line *ac* shown in Mr. Lessells' Fig. 1. By plotting the curve the performance is shown at the beginning of loading, which is the point most likely to err in the transverse test.

In regard to Mr. Lessells' remarks on Pearce's paper, it is obviously impossible to get the elastic deflection at failure; but as Mr. Pearce showed, it is possible by measuring the elastic deflection at two points, say at 50 per cent and 75 per cent of the maximum load, to obtain the deflection at 100 per cent load by extrapolation with considerable assurance. While this assumption is not strictly true, it is a valuable working approximation.

—JAMES T. MACKENZIE
American Cast Iron Pipe Company

To the Editor:

● Accurate values of elastic modulus both in tension and torsion are often necessary. The designer requires accurate values in calculations involving vibration phenomena. A recent publication, "Torsional Vibration," by W. A. Tuplin, published by John Wiley & Sons, lists such values as 17,000,000 lb. per sq.in. for tension and 6,700,000 lb. per sq.in. for torsion, but these are questionable values.

The writer does not take issue with the transverse tests as a control test, and so states in his article. However, some more accurate information than that given by such a test is necessary and the previous article showed how such could be obtained.

—JOHN M. LESSELLS
Swarthmore, Pa.

Fitting Equations to a Curve

C. J. SCHILLING
East Cleveland, Ohio

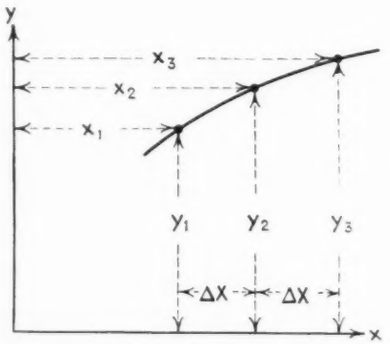
● With practical accuracy, a large variety of engineering data may be represented by a power equation of the form,

$$y = a + bx + cx^2 + dx^3$$

The process of fitting this equation to a curve representing the data is ordinarily a tedious operation requiring considerable time. When the coordinates of the known points taken along one axis are equally spaced this tedious procedure can be eliminated. Assuming equal spacing of the x-points, the equations given below give directly the constant for each of the terms. For the quadratic equation of the form given in the diagram these relations are obtained as follows:

Let the coordinates of the three points shown in the accompanying diagram be (x_1, y_1), (x_2, y_2) and (x_3, y_3) with the x-increments equal to m, a constant.

$$y_1 = a + bx_1 + cx_1^2$$
$$y_2 = a + bx_2 + cx_2^2$$
$$y_3 = a + bx_3 + cx_3^2$$



When the x-coordinates are equally spaced, constants for the power equation are given directly by the accompanying relations

By subtraction the term a can be eliminated:

$$y_2 - y_1 = b(x_2 - x_1) + c(x_2^2 - x_1^2)$$
$$y_3 - y_2 = b(x_2 - x_1) + c(x_3^2 - x_2^2)$$

In these two equations the right hand term involves values of x differing by the same amount since the x-increments m are made equal. Thus, this right hand term $(x_3^2 - x_1^2)$ can be written $2x_1 m + m^2$, the last term in the second equation above taking a similar form.

After making these substitutions the term b can be eliminated by subtraction giving the equation:

$$(y_3 - y_2) - (y_2 - y_1) = 2cm^2$$

From this equation the term c becomes:

$$c = \frac{y_3 - 2y_2 + y_1}{2m^2} \qquad (1)$$

With the value of c determined, values for b and a are then obtained as follows:

$$b = \frac{y_2 - y_1}{m} - c(2x_1 + m) \qquad (2)$$

$$a = y_1 - bx_1 - cx_1^2 \qquad (3)$$

In applying these equations to a given problem the constant c is found first from Equation (1) and this value is then substituted in Equation (2) to find constant b. Then, both c and b are substituted in Equation (3) to find a, the constant of the first term.

By a similar procedure the four constants in the cubic equation may be obtained by the formulas given below when the difference between successive x-coordinates are taken at equal intervals.

$$d = \frac{y_4 - 3y_3 + 3y_2 - y_1}{6m^3}$$

$$c = \frac{y_3 - 2y_2 + y_1}{2m^2} - d(3x_1 + 3m)$$

$$b = \frac{y_2 - y_1}{m} - c(2x_1 + m) - dK$$

where $K = (3x_1^2 + 3x_1 m + m^2)$

$$a = y_1 - bx_1 - cx_1^2 - dx_1^3$$

Swaging

To the Editor:

● In the first article by J. K. Olsen on "Stampings" in the December number is shown the design of a square driving shaft swaged from tubular stock. After mentioning that "the swelled out corners can be removed after swaging," the author has omitted one important detail, the lengthening of the shaft from swaging. The stock must go some place, some of it going to the corners which will be removed when the outside diameter is ground. From experiments we find that the shaft lengthens during swaging in proportion to the diameter of the stock. This factor must also be taken into consideration in such applications of swaging.

—C. A. UNDERWOOD
Waltham, Mass.

Standardization of Detailed Parts

E. R. HARRIS
Westinghouse Electric & Mfg. Co.

● Standardization is today the most important approach to greater industrial efficiency. Actual savings that are now being made in the automobile industry alone through organized standardization activities are estimated by the industry itself at millions of dollars a year.

By reducing types and sizes to a minimum without interfering with the design of the apparatus as a whole, considerable economies result. There are less kinds, but those that remain can be bought in larger quantities and consequently at lower prices. This can be done only when the designers of different pieces of apparatus are willing to cooperate in eliminating sizes that come close to each other. In reducing the number of sizes, a logical series should be established by having series arranged in uniform per-

centage increases, which leads to the use of "Preferred Numbers."

Greater interchangeability is a result of a smaller number of detail parts. Not only do the parts become fewer in number, but the reduction in numbers has its effect on the whole assembly. For instance, by standardizing on cold rolled round stock and by keeping the number of sizes to a workable minimum, the sizes or bores into which shafts or pins fit will be fewer. Consequently, there is a saving in reamers, gages, tools and fixtures, and cores for castings.

By keeping an eye on national standardization and on the suppliers' market, it is possible to buy parts from manufacturers who specialize in a certain detail and are able to supply it better, quicker, and cheaper than if it had to be made to each individual designer's opinion or fancy. This can be illustrated by the experience of one manufacturing concern that was making heat-treated nickel bolts to be used in a special application. The cost was rather high and after investigation a supplier was found who had this special bolt as one of its standard products. By the use of modern equipment and methods, this supplier furnished the bolt at one third of the former price.

Less storage space, bookkeeping, and handling are required, with the saving in attending overhead such as light, power and depreciation.

Following are typical examples of savings made possible by standardization in the plant of one manufacturer.

Subject	Original Number	Present Number	Approximate Savings per year
Steel Tubing	35	22	$700
Brass Tubing	206	61	5,700
Copper Tubing	78	21	2,500
Slate Bases	336	73	12,000
Soapstone	150	0	6,750
Cable	350	195	7,600
Oil Rings	72	44	7,200

Before standardization there were 2,436 different sleeve connectors in use. After standardization there were 155 items. Originally the sleeve connector was made to fit the application of a specific conductor. That is, each connector was made for the particular wire or cable, allowing a clearance over the conductor of from 0.005 to 0.015 in. The length of connector varied from $\frac{1}{4}$ to $2\frac{1}{2}$ in. depending on the size of conductor. When the workmen used the connectors they found them too small for conductors which had rough edges caused by the cutting-off operation, and the workmen would then have to get a larger size connector. A standard line of connectors with lengths of $\frac{3}{8}$ and $\frac{1}{2}$ in. was developed with a series of inside dimensions starting at 0.060 x 0.120 in. and advancing to 0.200 x 0.500 in. in regular increments. From these connectors the designer can select a size to fit the conductor allowing a specified clearance depending on the size of the copper.

Wooden handles have always given trouble in manufacturing, as many of them were split during assembly. A demand arose also for colored handles to be used in theatre lighting equipment, and a line of six handles was developed using molded composition in the desired colors. It was found later than the three smaller sizes would cover all requirements. The handles, averaging 70 per cent cheaper than the wooden type, can be used with a larger variety of studs and have a better appearance. Previously there were 134 handles on 55 drawings.

The total number of washers was formerly 1,329. These included various diameters and thicknesses. An analysis of these sizes showed that for a given size of bolt, screw, or stud it would be possible to have three sizes—standard, wide, and extra wide—discontinuing all intermediate sizes. The diameters were based partly on the preferred number series, and thicknesses were determined on a mechanical strength basis. Only 150 standard sizes are now used.

The designer by taking advantage of standardization of detail parts can help himself materially by having fewer parts, greater interchangeability, quicker service—all at a lower cost.

Standards for Helical Gears

F. A. MICKLE
University of Michigan

● British "Standard Specifications for Machine Cut Gears" can be compared with the "American Gear Manufacturers Association Recommended Practice" only on the basis of strength since the A.G.M.A. standard does not have a separate specification for surface stress and wear. Also, the British standard contains a factor based on speed reduction which is not present in the American standard. Thus, any comparison between the two standards must be calculated for several speed reductions.

In computing the horsepower per inch width of face for a helical gear with a helix angle of 30 deg. and a pressure angle of 20 deg., the British standard gives a bending strength factor of 22,000 lb. per sq.in. while the A.G.M.A. factor is 15,000 lb. per sq.in., assuming the material is high carbon or alloy steel with an elastic limit of 60,000 lb. per sq.in. As shown in the accompanying tabulation, the safe loads given by the two standards vary widely.

As the speed reduction increases, the capacity as computed by the British method increases and even at a 9:1 reduction is still smaller than the safe load given by the A.G.M.A. standards. As the speed increases, British values are subject to greater change than the loads given by American practice.

In the smaller gear sizes the British formula is more conservative, whereas in the larger sizes the A.G.M.A. method results in lower loads.

Differences between the results of the two methods, as shown in the accompanying tabulation, are too great. Perhaps neither standard is the correct one.

Comparison of British and American Standards for Helical Gears

Horsepower Per Inch Width of Face

Pitch Diam.	Pitch No.	Speed r.p.m.	Horsepower		
			British Standard		A.G.M.A.
			1.1*	9.1*	Any*
3	6	100	1.48	1.93	2.13
		900	8.60	11.15	15.14
		3600	23.97	31.07	48.04
3	12	100	1.04	1.18	1.28
		900	6.03	6.85	9.09
		3600	16.79	19.09	28.83
18	1	100	46.3	69.3	62.8
		900	310	401.0	395.0
		3600	863	1117.0	1088.0
18	4	100	22.6	.....	20.5
		900	130	.....	129.0
		3600	364	.....	356.0

*Gear Reduction

Roller Applications

ADAM FREDERICKS

In these twenty-three designs of roller mountings various phases to be considered are type, pressure, lubrication and ease of accessibility and adjustment. Both pin and roller should be ground, especially on the contacting surfaces, and the roller hole should be lapped

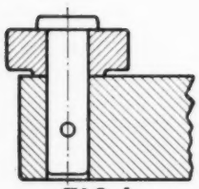


FIG. 1

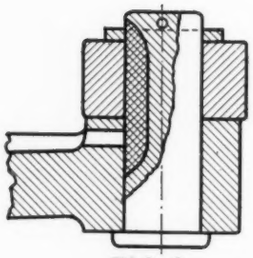


FIG. 2

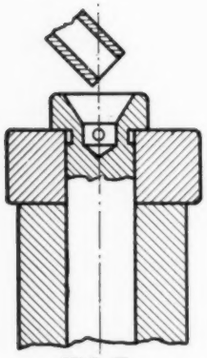


FIG. 3

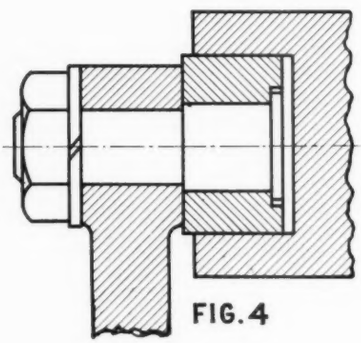


FIG. 4

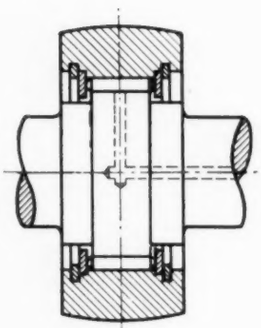


FIG. 5

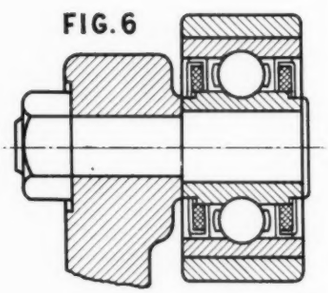


FIG. 6

Fig. 1—Head of pin retains overhanging roller. Bearing face relieved to reduce friction. Difficult to replace as two pins must be removed.

Fig. 2—Geneva wheel roller retained on stud by washer and cotter pin. Stud milled for felt, is lubricated from reservoir. Stud held in lever by press fit.

Fig. 3—Oil from overhead tube reaches roller through cross hole in stud coinciding with grinding relief.

Fig. 4—Countersunk roller retained by headed pin. Stud held by lock washer and nut. Splash lubricated.

Fig. 5—Large guide roller mounted on needle bearings retained by hardened and ground washers and snap rings.

Fig. 6—A shell pressed on single row sealed ball bearing, which is lubricated for life when manufactured. Headed retaining stud clamped in lever with hex nut.

Fig. 7—Needle bearing mounted roller retained on headed stud by snap spring retained washer. Zerk lubricated.

Fig. 8—Hex head bushing is relieved for needle rollers, which are retained by snap spring at **A**. Hex head screw fastens bushing to roller arm. Roller floats axially.

Fig. 9—Straddled roller mounting distributes load on two bearings. Roller runs on cottered pin. Straight pin pressed in roller is also good design.

Fig. 10—Straddling flanges retain needle rollers. Stud pinned on one flange. Rollers are held in place during assembly by short dummy pin.

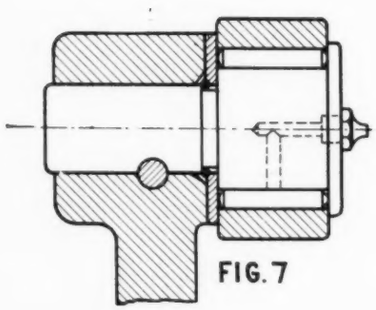


FIG. 7

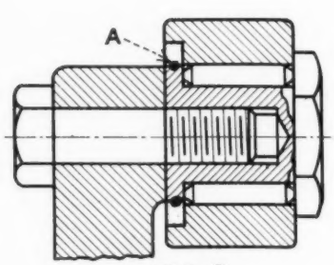


FIG. 8

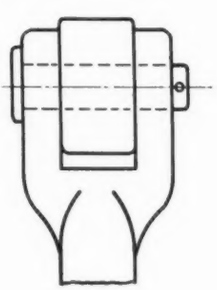


FIG. 9

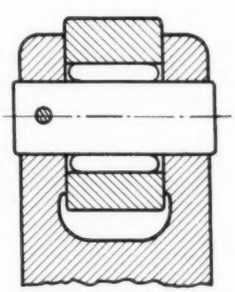


FIG. 10

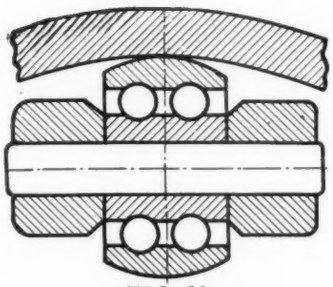


FIG. 11

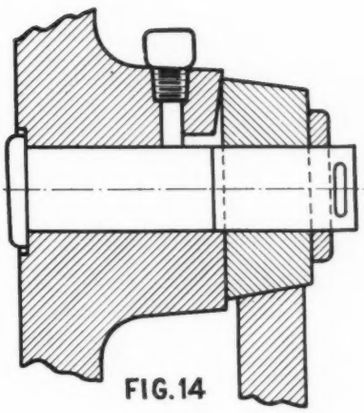


FIG. 14

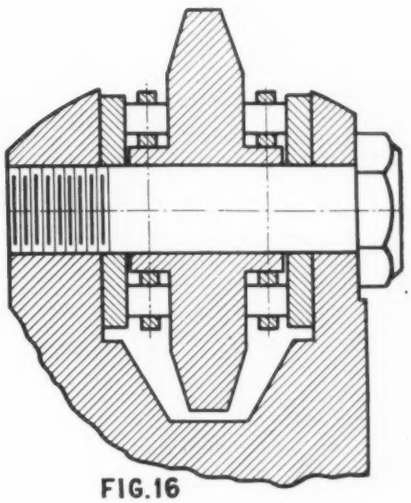


FIG. 16

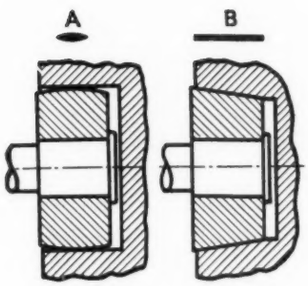


FIG. 12 FIG. 13

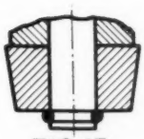


FIG. 15

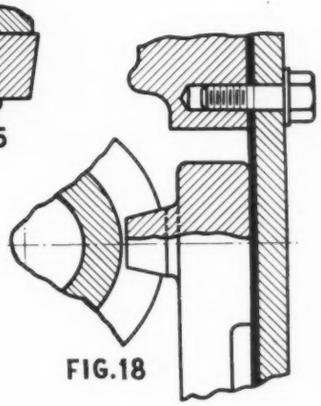


FIG. 18

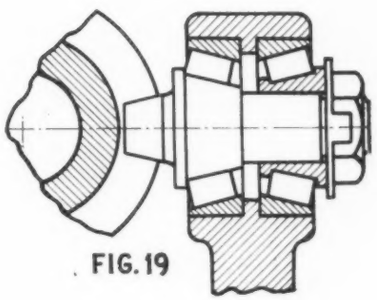


FIG. 19

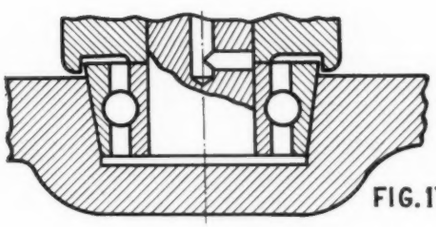


FIG. 17

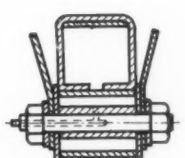


FIG. 21

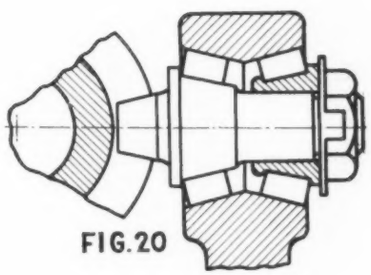


FIG. 20

Fig. 11—Special ball bearing has spherically ground outer race to engage concave surface. Pin pressed into inner race.

Fig. 12—Spherical faced cam rollers carry light loads only. At **A** is shown shape of contact area under load.

Fig. 13—With tapered cam rollers, contact area is rectangular, as at **B**, and greater loads can be carried.

Fig. 14—Tapered roller retained with washer and cotter. Bearing surface has triangular oil groove.

Fig. 15—Snap spring in pin groove for roller retention.

Fig. 16—Automobile steering roller with plain radial bearing and roller bearings for thrust loads.

Fig. 17—If outer race is not of sufficient thickness it will crack under load. Bearing should be guarded.

Fig. 18—Steering stud forged integrally with lever is adjusted by removal of shims.

Fig. 19—Another design for steering stud wherein the tapered roller bearing mounting allows pin to revolve.

Fig. 20—Variation of Fig. 19. Rollers contact hardened surfaces in lever, eliminating bearing outer races.

Fig. 21—Built up castor of pressed steel runs on needle bearings lubricated through center stud.

Fig. 22—The groove of a barrel cam acts as roller retainer.

Fig. 23—Ball bearing race grooved for round track. Outer race could be made half round to run in grooved track.

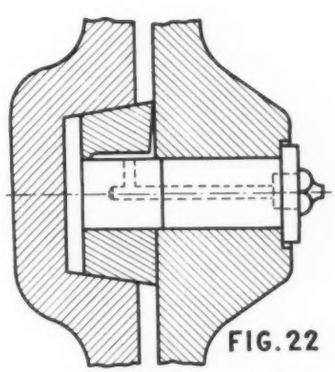


FIG. 22

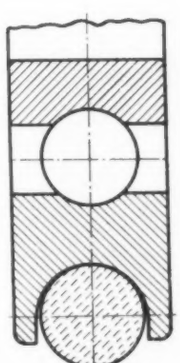


FIG. 23

NEWS

In the Coldspot refrigerator, a stamped head on the door completes the vertical stripe effect when the door is closed

Better Prospects Encourage Refrigerator Manufacturers

ELECTRICAL refrigeration proved to be depression-proof in 1934. Almost as many household electric refrigerators were sold last year as in the two boom years of 1928 and 1929 combined. For 1935 the Westinghouse company is reported to anticipate a 40 per cent increase in sales, and in preparation for a record-breaking year have brought out the new model shown here.

Retaining the same compressor unit, the cabinet and fittings have been redesigned, giving better insulation and increased convenience, while special attention to noise elimination makes it difficult to tell when the compressor is running. A revolving shelf, a door tray, and a die-cast ejecting ice cube tray are added conveniences. Dulux finish is used on the outside, enamel on the inside.

Several other manufacturers are also announcing new models in preparation for a banner year. To carry out their determination to offer an "outstanding design that will cause everyone to want it," Sears Roebuck and Company commissioned Raymond Loewy to create the cabinet design and plan the interior of their 1935 refrigerator.

New insulation, new finish, noise elimination and a new interior arrangement are featured in the latest Westinghouse refrigerator

Triple ribs can be seen above and below the door opening in the accompanying interior view of the Coldspot. Matching vertical ribs on the outside of the door form an unbroken vertical stripe when the door is closed, these ribs also serving as stiffeners for the panels.

Perhaps the most unusual feature of the Coldspot is the door operating mechanism. A slight push on the vertical bar opens the latch, and the door is swung open about 25 deg. by a spring pin on the hinge side of the door frame. Decoration on the latch bar, hinges and nameplate duplicates the vertical stripe effect. All corners are rounded and the outside of the box is free of projections, angles or creases. This cabinet is built by the Seeger Refrigerator Company.

The rotary compressor and refrigerator used in the Coldspot are furnished by the Sunbeam Electric Manufacturing Company. The unit, placed at the top of the cabinet, has only five moving parts.

Cooling coil is tinned copper. Interior decorative trim is anodic-treated aluminum and carries out the same motif as the exterior of the cabinet. Wire shelves are tin coated and are supported on Plaskon pegs extending from the sides.

The cabinet is insulated with Dry Zero, and Micarta strips over the door edges, together with rubber strips, seal the door. Inside and outside finish is baked enamel for all except the lower-priced models, which have Dulux outside finish.

Easy Installation for Easier Selling

WORKING on the theory that much sales resistance would be overcome if the customer were not burdened with the expense of installation of electric fixtures by an electrician, a manufacturer recently introduced a line of replacement electric lighting fixtures that can be installed as easily as putting a new bulb in a socket.

In the design of these new lighting fixtures, a standard lamp base thread is provided so that the fixture can be screwed into a standard receptacle. Thus for houses that have once been equipped with receptacles for this type of fixture, it is merely necessary to unscrew the fixture and put in the new one, in exactly the same manner as a light bulb would be replaced.

Spot Welding Record Without Failure

IN CHECKING their production records, the General Electric Company reports that they have produced as many as six million spot welds and 1,500,000 in. of seam welding on the evaporator units of the G.E. refrig-

erator without a single reject because of faulty welds. This record is largely attributed to the automatic operation of their spot-welding machines which are equipped with Thyratron control.

In the General Electric refrigerator the evaporator unit is made of stainless steel, each one requiring 310 spot welds and 75 in. of seam welding. The spot welding is carried out at the rate of 150 welds per min. and the seam welding at the rate of 72 in. per min. It is expected that as a result of developments now taking place even these high speeds will be increased. The aim is seam welding at approximately 100 in. per min. and spot welding at a correspondingly higher rate.

Decimal Dimensioning Adopted by Ford

AS reported by Athel F. Denham in the December 15th number of *Automotive Industries,* the Ford Motor Company has adopted the decimal system to replace the fractional system for dimensioning. Decimals, as compared to common fractions, provide some of the advantages claimed for the metric system.

With the decimal system as adopted by the Ford Motor Company, addition and subtraction of dimensions is greatly simplified. Other engineering calculations are likewise made much easier and arithmetical errors are less likely. For example, calculations for sines, cosines and other angle functions are much easier. In fact, wherever multiplication or division is involved, the decimal system saves much time and labor.

In the system as adopted by Ford dimensional tolerances are indicated by specifying the limiting dimensions. The decimal fractions or the number of digits to the right of the decimal point, where no limiting dimensions are specified, indicate the degree of accuracy required. To specify an increasing degree of required machining accuracy, additional digits are added to the right of the decimal point.

It is claimed that the changeover from the common fraction system to the decimal system can be made readily without any great confusion or involving any great expense.

New Journal of Applied Mechanics

BEGINNING March, 1935, the Applied Mechanics Division of the American Society of Mechanical Engineers will publish quarterly a *Journal of Applied Mechanics.* The technical editor will be J. M. Lessells.

According to the announcement, the journal will publish original papers in general mechanics, elasticity, hydrodynamics, aerodynamics, strength of materials and thermodynamics, similar to those published during the last few years in the *Transactions of the Applied Mechanics Division* of the society. Book reviews and progress reports will be included.

For non-members of the society the annual subscription price will be $5. Subscriptions may be sent to A.S.M.E. Applied Mechanics Journal, 29 West 39th Street, New York.

New Laminated Sheets

Cloth, plastics, wood and paper can now be bonded on steel

SHEETS of steel with a cloth facing bonded to the steel, thin veneers of wood bonded to steel, paper bonded on the steel and synthetic resin sheets bonded to steel were demonstrated by Howard W. Blakeslee, Science Editor of the Associated Press, in a meeting of the Science Forum of the New York Electrical Society on January 15.

These bonded sheets are the latest further development of the Mellon Institute, the H. H. Robertson Company and the Mica Insulator Company in their efforts to devise methods of producing such laminated sheets, as reported on p. 313, August, 1933, *P.E.* The steel sheets are first coated with a special zinc alloy developed by the Mellon Institute. This alloy merges with both the steel and the cladding material, thereby effecting a one-piece sheet.

Further developments in bonding paper sheets to steel led to photographed finishes. Sheets of sensitized paper are photographically printed to give the desired effect. The paper print is then bonded to the steel sheet. Thus, the effect of a mahogany wood finish, or any other desired surface finish, can be obtained readily.

MEETINGS

American Institute of Mining and Metallurgical Engineers — Institute of Metals Division, Engineering Societies Bldg., New York, N. Y., Feb. 18-21. A. B. Parsons, secretary, 29 West 39th St., New York, N. Y.

EXHIBITIONS

International Trade Fair—Grand Central Palace, New York, N. Y., Feb. 4-9. International Trade Fair Inc., 480 Lexington Ave., New York, N. Y.

Industrial Arts Exposition—Rockefeller Forum, 30 Rockefeller Plaza, New York, N. Y., April 15-May 15. T. J. Maloney, manager, 386 Fourth Ave., New York, N. Y.

Improved Synthetic Rubber

ALTHOUGH at present the prohibitive cost of the material precludes its adoption as a general substitute for rubber, the remarkable properties of the new synthetic rubber-like material developed by The B. F. Goodrich Company will undoubtedly find certain specialized applications.

This new material, which has been named "Koroseal," is in many respects similar to rubber, although it is not the same as rubber in chemical composition. Like rubber, the properties of the material can be varied by a proper selection of the compounding methods. Thus, the material may be made extremely hard or soft, or it can be given a doughy consistency to make it possible to mold it into any shape. The material is also produced in a variety of colors, and is odorless.

Unlike rubber, Koroseal is much more resistant to swelling when exposed to oils and greases. It also to a great extent resists disintegration from corrosive chemicals, being resistant to the action of chromic acid and hot nitric acid. One suggested application of the material has been for piston packing because of the resistance of the material to deteriorating effects from oil. Another feature of this new material is its great resistance to light and to oxidation.

Resistance to moisture and stability over a wide temperature range make soft Koroseal ideal for molding ornamental plaster casts. Over 500 casts have been made in Koroseal molds still in service.

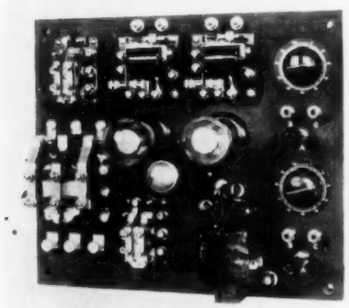
It is not expected that Koroseal will replace rubber for general use, but it is expected that this new material will find increasing applications where physical properties like those of rubber combined with superior resistance to oils and chemicals or to repeated flexing, are required.

New Materials and Parts

Hydraulic Cylinders

Are double-acting piston ring type for high pressure hydraulic service. Longitudinal tie rods have been eliminated, which facilitate replacement of gaskets and renders accessibility to the internal parts. The caps may be assembled independently of each other so for which reason the inlet and outlet ports can be located at top, bottom, or either side. Standard models are available for working pressures up to 1,500 lb. per sq.in. and special cylinders can be furnished for higher pressures and any length of stroke. Hannifin Mfg. Co., 621-631 S. Kolmar Ave., Chicago, Ill.

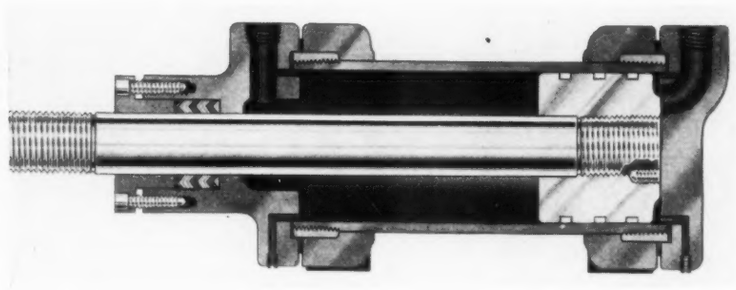
Automatic Repeat Process Timer

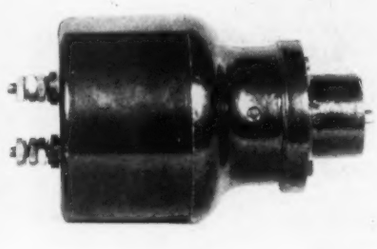


For use with a variety of electrically operated devices and machines requiring automatic timing of "power-on" and "power-off" periods, which are independently adjustable in a range of 0.05 to 0.9 sec. When both are set at their minimum, the smallest timer will provide 550 operations per minute. On the largest timer the range is 0.12 to 7.5 sec. Photograph shows timer removed from its inclosing cabinet. Four standard sizes are obtainable for operation on 110, 220, 440 and 550 volt circuits of any standard frequency. The Electrical Controller & Mfg. Co., 2700 E. 79th St., Cleveland, Ohio.

Portable Potentiometer

Used for checking thermocouples, millivoltmeters and potentiometers quickly and accurately. With the combination of a step-switch and a slide wire, which is equivalent to having a slide wire 109 in. long, makes it possible to read the indicating scale to 0.01 millivolts over a total range of 71 millivolts. The indicating scale is 17 in. long, making the smallest scale division of 0.05 millivolts over $\frac{1}{16}$ in. in width. Two sets of binding posts and a toggle switch provided for thermocouple checking. The circuit draws 3 milliamperes from a dry cell. The case is $9\frac{3}{8}$x$10\frac{1}{4}$x$6\frac{1}{4}$ in. and weighs 12 lb. The Brown Instrument Co., Philadelphia, Pa.





Quick Reading Tachometer

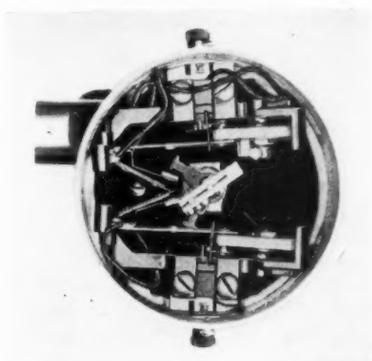
Made especially for recording engine speeds of aircraft or motor boats, but applicable to any equipment which can be provided with SAE threaded outlets. It operates on a generated-frequency principle and the rotor is direct driven, requiring no gearing. Voltage loss or polarity of leads are said not to affect accuracy of reading. Dial graduated from 0 to 2,500 r.p.m. when generating unit turns 0 to 1,250 r.p.m. General Electric Co., Schenectady, N. Y.

Purolator Oil and Coolant Filters

For industrial uses are placed in the coolant pipe line to remove all foreign matter. These "Purolators" are available in several designs, one of a cloth element type, having several hundred sq.in. of filter cloth. The metal type has a cylinder of wire, wound in slots 0.015 to 0.0005 in. wide. In the larger sizes, stationary knives are used and the element is motor driven at the rate of 3 r.p.m. In some cases, 3 degrees of filtration are provided through cloth, metal, and

again through cloth. Other materials may be filtered, such as greases, compressed air, gasoline, paint, varnishes and lacquers, refrigerating fluids, waxes, syrups, and tooth paste. Motor Improvements, Inc., 365 Frelinghuysen Ave., Newark, N. J.

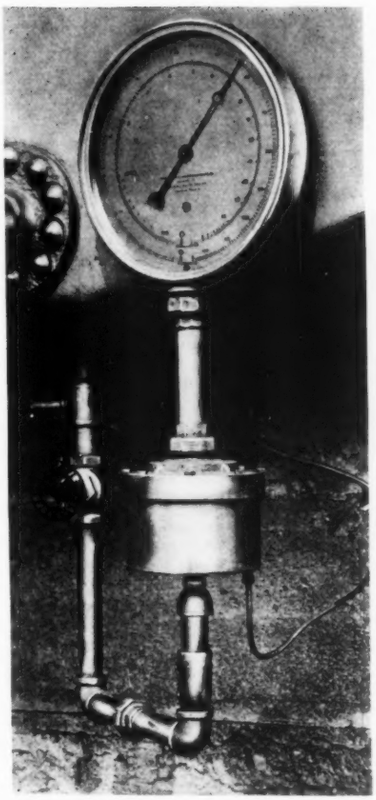
Fan and Limit Switch



For high voltage fan and furnace control of two wire type. Fan switch closes circuit upon temperature increase thereby starting fan to circulate air when the predetermined temperature setting is reached. Limit switch, actuated by a bi-metal element, breaks circuit when desired furnace bonnet or plenum chamber temperature is reached. Range of fan setting 50 deg. to 400 deg. F. Range for limit setting 100 deg. to 400 deg. F. Furnished for high voltage or with binding posts for low voltage or both. Fan motor ratings from $\frac{1}{4}$ to $\frac{1}{2}$ hp., 110 to 220 volts a.c. and d.c. current. Detroit Lubricator Co., Detroit, Mich.

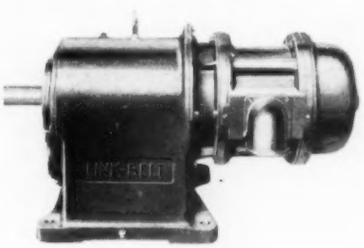
Liquid Gage

Hydrostatically measures amount of fluid in a tank. Liquid is connected by means of a copper tube to a bellows pipe and metal diaphragm, which is linked mechanically to the indicator of the gage. As the tank is filled the diaphragm expands, moving the indicator pointer. Dial reading shows the quantity in terms of gallons, pounds, barrels, inches, or any other desired unit. Gages are built in capacities from 300 to 3,200,000 gallons. Imperial Electric Co., Akron, Ohio.



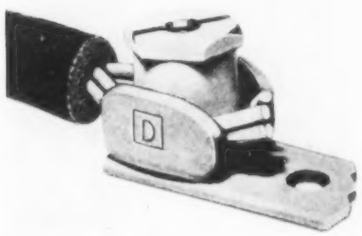
Motorized Speed Reducer

Is driven by a standard round frame motor, which may be removed for inspection or maintenance without disturbing alignment of motor or gears. Heat-treated helical gears are mounted on anti-friction bearings incased in a gray iron housing. Lubrication of gears and bearings is automatic and bearing seals are provided at the shaft openings. May be mounted in any position and are available in double reduction for $\frac{1}{2}$ to 75 hp. in ratios up to $38\frac{1}{2}$ to 1, and in triple reduction up to 30 hp. in ratios of 292 to 1. Link-Belt Co., 910 S. Michigan Ave., Chicago, Ill.



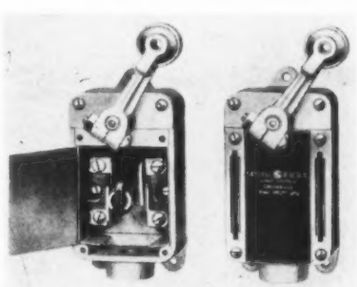
Solderless Connector

Consists of a copper cup with inclosing sides and raised lip of curved contour at the wire entrance, a hex nut and stud. Nut may be turned with screw driver or wrench. Said to be vibrationless and fits in same space as a solder lug. Available for No. 14 to No. 4 wire for 60 amp. size and No. 14 to No. 1 wire for 100 amp. size. Square D Co., Detroit, Mich.



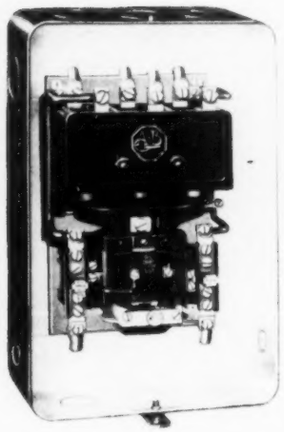
Limit Switch-Track Type

Small spring-return switch for making and breaking control or indicating circuits at a fixed point in the travel of the part of a machine or mechanism. Has one normally open and one normally closed circuit and is operated by 18 deg. movement of arm and overtraveled 54 deg. without damage. Contacts are of the double-break type, silver plated. Inclosed in die-cast case $4 \times 2\frac{7}{16} \times 1\frac{23}{32}$ in. in depth. Maximum capacity is 5 amp., 550 volts a.c. and from 0.4 amp., 550 volts to 2 amps. at 115 volts d.c. General Electric Co., Schenectady, N. Y.



Solenoid Switch

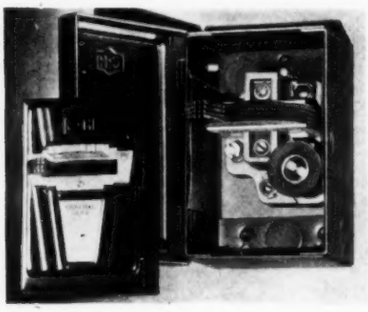
An arc hood completely incloses each contactor in individual compartment from which arcing cannot escape. Contacts are made of silver alloy. The pick-up and drop-out voltages are said to be low, closing at 70 per cent of rated voltage and will not drop open unless voltage falls below 50 per cent of normal. Each



switch unit is insulated and can be attached to machine without further insulation. Thermal relays protect against overloads. Available in 50 amp. and 100 amp. ratings and will handle polyphase motors up to 15 hp., 110 volts; 30 hp., 220 volts; and 50 hp., 440-550 volts, or can be used for starting single phase motors that may be thrown across the line. Allen-Bradley Co., 1311 S. First St., Milwaukee, Wis.

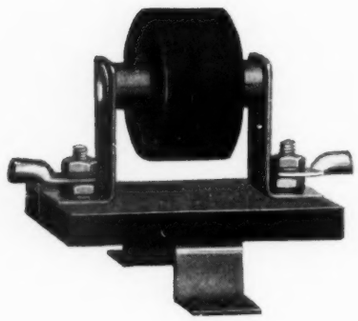
Safety Switch

Used for oil burners, air conditioning equipment and electric unit heaters. Is of fusible type, toggle front operated, double pole for positive make and break. Incased in black baked enamel box having cadmium trim, and with knockout plug at top, bottom, side and rear. Switch will accommodate 30 amp. Cutler-Hammer, Inc., 12th & Paul Ave., Milwaukee, Wis.



Insulation Protector

Using the same principle as the "Thyrite" lightning arrester, discharge arrester is designed to protect insulation of d.c. magnetic circuits. Resistance material is of inorganic, ceramic material, hard as marble, which conducts about sixteen times more current each time the voltage is doubled. Practically an insulator at line voltages, it reduces continuous watt loss to approximately 2 per cent of that for equivalent fixed resistors. Designated as Type CR9196, "Thyrite" discharge resistors, units are available in two ratings—one for 110 to 275 volts; the other for 500 to 650 volts. General Electric Co., Schenectady, N. Y.



Bearing Material

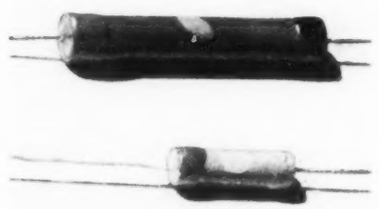
Used primarily in the paper industry and in wood-working plants, this bearing material is made of a compound of asbestos, graphite and rubber and is known as "Eel-Slip." Said to be tough and readily machined, having a low coefficient of friction even though unlubricated. Water is its best lubricant. May be run continuously submerged in water. Johns-Manville Co., 22 East 40th St., New York, N. Y.

Blueprint Paper and Cloth

For quick printing, a "non-bleeding" paper and cloth is said to be color-fast, though over-exposed, and to give a deep blue background and sharp white line without any fading tendency. Available in all standard lengths, widths and weights and rag stock content, in rolled or cut sheets for both white and blue line printing. The C. F. Pease Co., 813-821 N. Franklin St., Chicago, Ill.

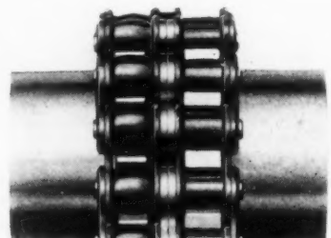
Extruded Resistor

Composed of a mixture of two materials, one of high resistance and one of low resistance. By varying the proportions of the two materials, various resistivities are obtained. Because of the material properties, the entire cross section is said to carry current with no voids, point contacts or variations in composition, giving longer service life and higher watt ratings without resistance variation. Leads are pigtails fastened to metal end caps. A range of sizes and capacities are available. Henry L. Crowley & Co., 1 Central Ave., West Orange, N. J.



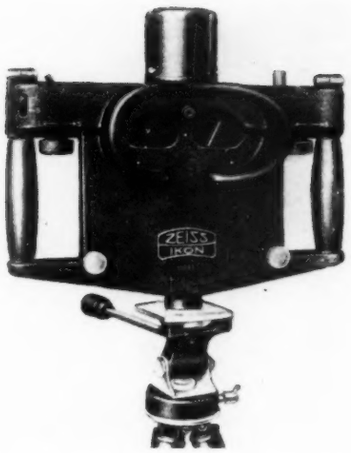
Flexible Coupling

Utilizes a special double width roller chain, one strand being equipped with standard cylindrical rollers and the other with convex surface over-size rollers. These convex rollers are said to provide for free operation with shaft deflection and misalignment up to 3 deg. The heat-treated rollers engage hardened sprockets. Baldwin-Duckworth Chain Corp., Springfield, Mass.



Stroboscope

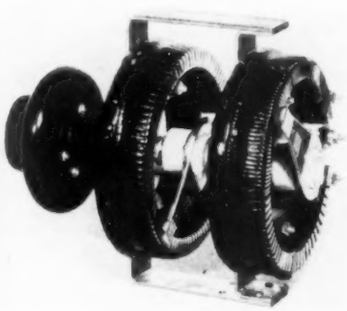
For studying rapidly moving machine parts, the part in motion is observed through a rotating slit disk whose speed can be regulated to coincide with the speed of the moving ob-



ject. Built-in speed indicator also serves as a tachometer for measuring speeds up to 140,000 r.p.m. without contact with the rotating part. The stroboscope can also be equipped with binoculars to obtain a closer view. The outfit is portable or may be used on a tripod. George Scherr Co., 128 Lafayette St., New York, N. Y.

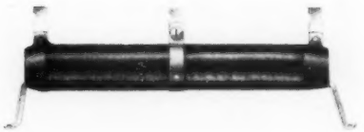
Concentrically Controlled Rheostats

Are mounted in tandem and only require space of a single unit. They may be adjusted independently by means of dual control knobs at the front. Large knob controls front rheostat and small knob, the rear rheostat. Shafts and bushings are made of porcelain so they may be mounted directly on machines. Available for any application within their watt rating range. Ohmite Mfg. Co., 636 N. Albany Avenue, Chicago, Ill.



Adjustohm Resistor

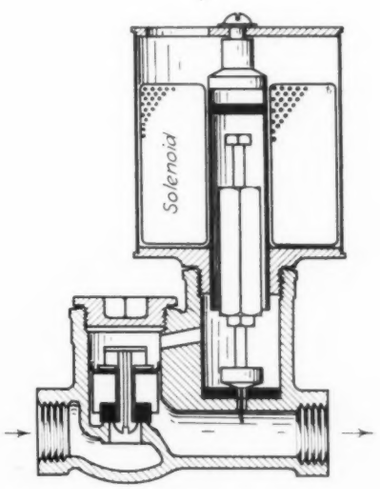
Is wound with a narrow strip of wire and vitreous enameled. Approximately ⅓ of the surface area is bared for contact between the wire and the adjustable band contact. Terminals



permit soldering flexible leads. Adjustment of the resistance is accomplished by loosening a scroll on the band. Units are available in resistance from 1 to 100,000 ohms, with ratings from 20 to 160 watts. Ward Leonard Electric Co., Mt. Vernon, N. Y.

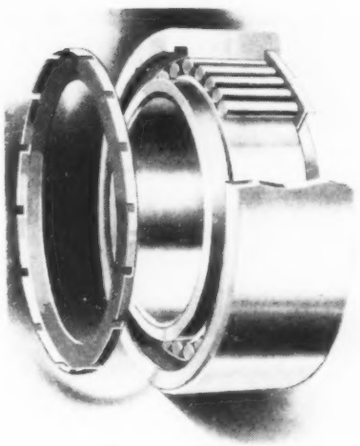
Plunger Type Valve

Solenoid operated for use on Freon, oil, air, ammonia and other liquids. Line pressure opens valve. Hydraulic pressure, three times greater than line pressure, holds valves closed. Valve will also close on equalized pressure. No gaskets, springs or levers required.



Made in sizes of ¼ in. to 3 in. Three-way valves using combination of two hydraulic valves made in single casting also available. A. F. Hoppe Engineering Inc., Indianapolis, Ind.

"Multirol" Roller Bearing



Has a full complement of small needle type rollers retained by hardened steel end washers which hold the raceways together as a unit. They also retain lubricant and exclude foreign material. Rollers and raceways are of hardened high carbon, chrome steel. Available in stock sizes from ⅞ in. to 4 in. bore with either single or double rows of rollers. They can also be supplied with an outer raceway and roller assembly only or with inner raceways and roller assembly only. McGill Mfg. Co., Valparaiso, Ind.

MANUFACTURERS' PUBLICATIONS

Lacquer—Roxalin Flexible Lacquer Co., Elizabeth, N. J. Booklet, 13 pages, "How Good Are You At Twisting and Bending" with steel, brass and aluminum samples, finished with one coat of lacquer.

Magnetic Clutch—Magnetic Manufacturing Co., Milwaukee, Wis. Bulletin No. 200, 4 pages, 8½x11 in., explaining the "Stearns High-Duty" multiple-disk magnetic clutch, showing application photographs and cross-section line cuts of type C clutches, 6 to 10½ in. diameter and type B, 22 to 46 in. diameters.

Motor Price Wheel—The Louis Allis Co., Milwaukee, Wis., consists of two fiber disks, 8 in. diameter, printed in colors, which quickly tells frame sizes and list prices of 448 popular sizes and types of electric motors.

"Leather Belting"—Chicago Belting Co., 113-125 N. Green St., Chicago, Ill. One-page circular describing and illustrating short center belt drives with spring tension adjustment.

Motors—Allis Chalmers Manufacturing Co., Milwaukee, Wis. Pamphlet 8½x11 in., explaining "Seal-Clad" motors with molded Bakelite shields to protect windings against moisture, metallic dust and oil. Built in ratings up to 25 hp., 1,800 r.p.m. with ball or sleeve type bearings. Assembled and disassembled views are shown.

Motors—The Ohio Electric Mfg. Co., 5900 Maurice Ave., Cleveland, Ohio. Bulletin No. 214-A, "Ohio Motors for

Air Conditioned Heating System," 4 pages, 8½x11 in., showing application of ⅛ to ¾ hp., a.c. and d.c. motors to air-conditioned heating systems with photographs and mounting dimensions.

Paint—Sherwin-Williams Co., 101 Prospect Ave., N. W., Cleveland, Ohio, has issued the first number of Vol. 1 of their new house organ entitled "Paint," devoted to paint, painting problems and painting practices. Stressing that "paint is a tool as surely as your loom or your lathe," and that "lighting costs are cut, rejections are reduced, safety is enhanced and vision—the most important factor in industry—functions at its best in a well-painted plant," this magazine will strive to show how to use paint as a medium to lower costs.

On the center pages, completely outside the field of paint, are shown pictures of such subjects as the editor feels will be of interest to the reader. They are representative of America's best photographers. The editor has in mind that "A thousand words are still a meager summary of a good picture." This first issue has eight pages and is 8½x11 in. in size.

Renewable Gate Valves—The Fairbanks Co., 393-399 Lafayette St., New York, N. Y. Folder 6x9 in., describing gate valves with renewable bronze seat rings, to facilitate repairs without removing body from pipe lines. For pressures up to 300 lb. per sq.in. in sizes of ½ to 16 in. diameters.

Sheet Iron—Republic Steel Corp., Massillon, Ohio. Booklet, "Sheet Iron—A Primer" 64 pages, 5½x7¾ in. Tells of the early history of iron and various processes of refining, rolling and treating. Gives physical properties and gage tables. Contains illustrations and diagrams of manufacturing methods and photographs of applications.

Steels—Union Drawn Steel Co., Massillon, Ohio. Pamphlet, "Important Features of Cold Drawn Steel" giving physical properties and tolerances. Illustrated.

Stop Nuts—Elastic Stop Nut Corp., P. O. Box 38, Elizabeth, N. J. Booklet, 12 pages, 8½x11 in. "Elastic Stop," describing the design of a lock nut having an inserted fiber collar which binds on the thread flanks. Illustrated with application photographs and tables of standard and special shaped nuts having coarse and fine threads made of steel and brass.

•

"Multiples At a Glance"—Timken Steel & Tube Co., Canton, Ohio. 6 pages of 8½x11 in. Charts of multiples up to 50 horizontally and 1 to 60 in. in length advancing in 1/64 in. vertically. One page devoted to properties of SAE and special steels.

BOOKS AND BULLETINS

The Alloys of Iron and Copper

J. L. Gregg and B. N. Daniloff. 454 pages, 6x9 in. Blue clothboard covers. Published by McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $5.

This is the fourth volume of the Alloys of Iron Research Monograph Series prepared at Battelle Memorial Institute. It is written in a concise but comprehensive manner from 634 pertinent articles and essential information from over 2,000 journals and text books in many languages. The alloying of iron and steel with copper are covered in 12 chapters, illustrated with photographs of test samples and microphotographs and with 110 various tables and 182 graphs and diagrams. The constitution of iron-copper and carbon alloys are explained in three chapters. Another covers the melting and working of copper-irons and steels showing the effect of hot and cold working, weldability and the effect of copper on pickling action. Effects are given of copper at critical points of ordinary and elevated temperatures, physical properties, carburizing, nitriding properties and hardening. Corrosion resistance for atmospheric, submerged and underground installations and protective coatings are discussed.

Introduction to Mechanics and Heat

Nathaniel H. Frank. 339 pages, 6x9 in. Red clothboard covers. Published by McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $3.

A treatment of the fundamental principles of mechanics and heat required as part of a two-year course in physics at the Massachusetts Institute of Technology. The book is divided into 22 chapters in which are introduced linear and plane kinematics and dynamics, statics of rigid bodies and elasticity, dynamics of rigid bodies, fluids and elasticity and the first and second laws of thermodynamics. Heat conduction, kinetic theory of ideal gases and properties of real gases are also expounded. Numerous problems follow at the end of each chapter. The reader should have an elementary knowledge of calculus.

Outdoor Weathering of Metals and Metallic Coatings

113 pages, 6x9 in. Published by American Society for Testing Materials, 260 South Broad St., Philadelphia, Pa. Price, $1.25 in heavy paper cover; $1.50 in clothboard covers.

Reprint of five technical papers: Outdoor Test Results on Bare and Metal-Coated Ferrous Specimens; The Harmony of Outdoor Weathering Tests; Influence of Rainfall and Smoke on the Corrosion of Iron and Steel; Early Interpretation of Test Results in the Atmospheric Corrosion of Non-Ferrous Metals and Alloys; Galvanic Corrosion by Contact of Dissimilar Metals.

Discussions contributed to each paper are included, along with a general discussion at the close of the book. Tables and charts presented with papers are included in the reprint.

•

Tin Research

Four recent reports, listed below, are published by the International Tin Research and Development Council. Copies are available free from L. J. Tavener, 149 Broadway, New York, N. Y.

"The Electrochemical Behavior of the Tin-Iron Couple in Dilute Acid Media," by *T. P. Hoar*, 10 pages, suggests that changes in the composition of the iron base sheet or additions to the corroding medium will reduce corrosion in tin plate.

"Tests on Tin-Base and Lead-Base Bearing Metals," by *C. Jakeman* and *Guy Barr*, 23 pages, concludes that lead-alkali metals alloys, give friction loss than 80 per cent tin alloys but are unsuited for load exceeding 900 lb. per sq.in. of projected bearing area.

"Behavior of Bearing Metals When Subjected to Various Deformation Tests," 38 pages, reports indentation tests, tensile properties and the results of pounding tests for several white metal bearing alloys.

"Beneficial Use of Tin Compounds in Lubricants," by *E. J. W. Mardles*, 4 pages, indicates that tin compounds reduce oxidation and change the viscosity of the lubricating oil.

Permissible Stress Range for Small Helical Springs

University of Michigan, *Engineering Research Bulletin No. 26*, by F. P. Zimmerli, 80 pages, $1.

For a given spring steel the endurance limit or stress range in torsion is reported to be directly proportional to the ultimate torsional strength. Specimens tested included 14 compositions, some with several heat treatments. Chrome vanadium steels showed the highest stress range of any of the steels tested.

Short Center Belt Drives

Calculations for the arc of contact and length of belts having an idler pulley

E. N. KEMLER
Assistant Professor of Mechanical Engineering
University of Pittsburgh

WHEN an idler pulley is used to increase the arc of belt contact on the driving pulley, it becomes necessary to calculate that increase to obtain the belt length. In Fig. 1 center lines are drawn connecting the three pulley centers.

Solving for the belt wrap θ on pulley d:

$$X_1 + X_2 = \sqrt{A^2 + B^2} = \frac{d + D_2}{2 \sin (\phi + \Delta)}$$

$$X_1 = d/2 \sin (\phi + \Delta)$$

$$X_2 = D_2/2 \sin (\phi + \Delta)$$

$$\text{Sin} (\phi + \Delta) = (d + D_2)/2\sqrt{A^2 + B^2}$$

$$\phi = \sin^{-1} [(d + D_2)/2\sqrt{A^2 + B^2}] - \Delta$$

$$\Delta = \sin^{-1} \frac{A}{\sqrt{A^2 + B^2}}$$

$$\phi = \sin^{-1} \frac{d + D_2}{2\sqrt{A^2 + B^2}} - \sin^{-1} \frac{A}{\sqrt{A^2 + B^2}}$$

The angle of belt contact on the driving pulley d will then be:

$$\theta = 180 \text{ deg.} - \alpha + (\phi + \Delta) \pm \Delta$$
$$= 180 \text{ deg.} - \alpha + \phi$$

In which the angle of approach α is

$$\sin \alpha = \frac{D - d}{2C} \text{ or } \alpha = \sin^{-1} \frac{D - d}{2C}$$

The angles ϕ and Δ can be found on the chart. When A is above the centerline, angle Δ will be minus and if A is below the centerline, angle Δ will be plus. The scale A in the chart can be used for either plus or minus values but the sign preceding the angle Δ must be kept in mind. When values of A are less than 1, values of angle Δ must be interpolated. For example, when A is between ± 0.5 in., angle Δ is less than ± 2 deg. and is read on the scales "A" and "angle Δ in deg." by interpolating.

For the example shown on the chart on the next page, the arc of belt contact on pulley d will be:

$$= 180 \text{ deg.} - \alpha + (\phi + \Delta) - \Delta$$
$$= 180 \text{ deg.} - 14.5 \text{ deg.} + 33 \text{ deg.} - (-4.5 \text{ deg.})$$
$$= 203 \text{ deg.}$$

Equation for the length L of belt is:

$$L = E + \theta + F + G + H + J$$

In which $E = \dfrac{D}{2} \dfrac{(180 \text{ deg.} + \alpha + \psi)}{57.3}$

$$\theta = \frac{d}{2} \frac{(180 \text{ deg.} - \alpha + \phi)}{57.3}$$

$$F = \frac{D_2}{2} \frac{(\psi + \phi)}{57.3}$$

$$G = C \cos \alpha$$

$$H = D \tan (90 \text{ deg.} - \psi + \lambda)$$

$$J = d \tan (90 \text{ deg.} - \phi + \gamma)$$

In the foregoing equations, values for the various symbols are calculated as follows:

$$\psi = \sin^{-1} \frac{D + d}{2\sqrt{A + (C - B)^2}}$$

$$\alpha = \sin^{-1} \frac{D - d}{2C}$$

$$\phi = \sin^{-1} \frac{d + D_2}{2\sqrt{A^2 + B^2}} - \tan^{-1} \frac{A}{B}$$

$$\lambda = \tan^{-1} \left(\frac{D_2/2 \cos \psi - A}{(C-B) - D_2/2 \sin \psi} \right)$$

$$\gamma = \sin^{-1} \left(\frac{D_2/2 \cos \phi - A}{B - D_2/2 \sin \phi} \right)$$

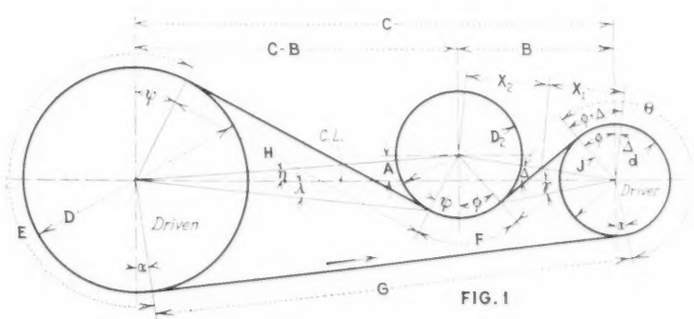
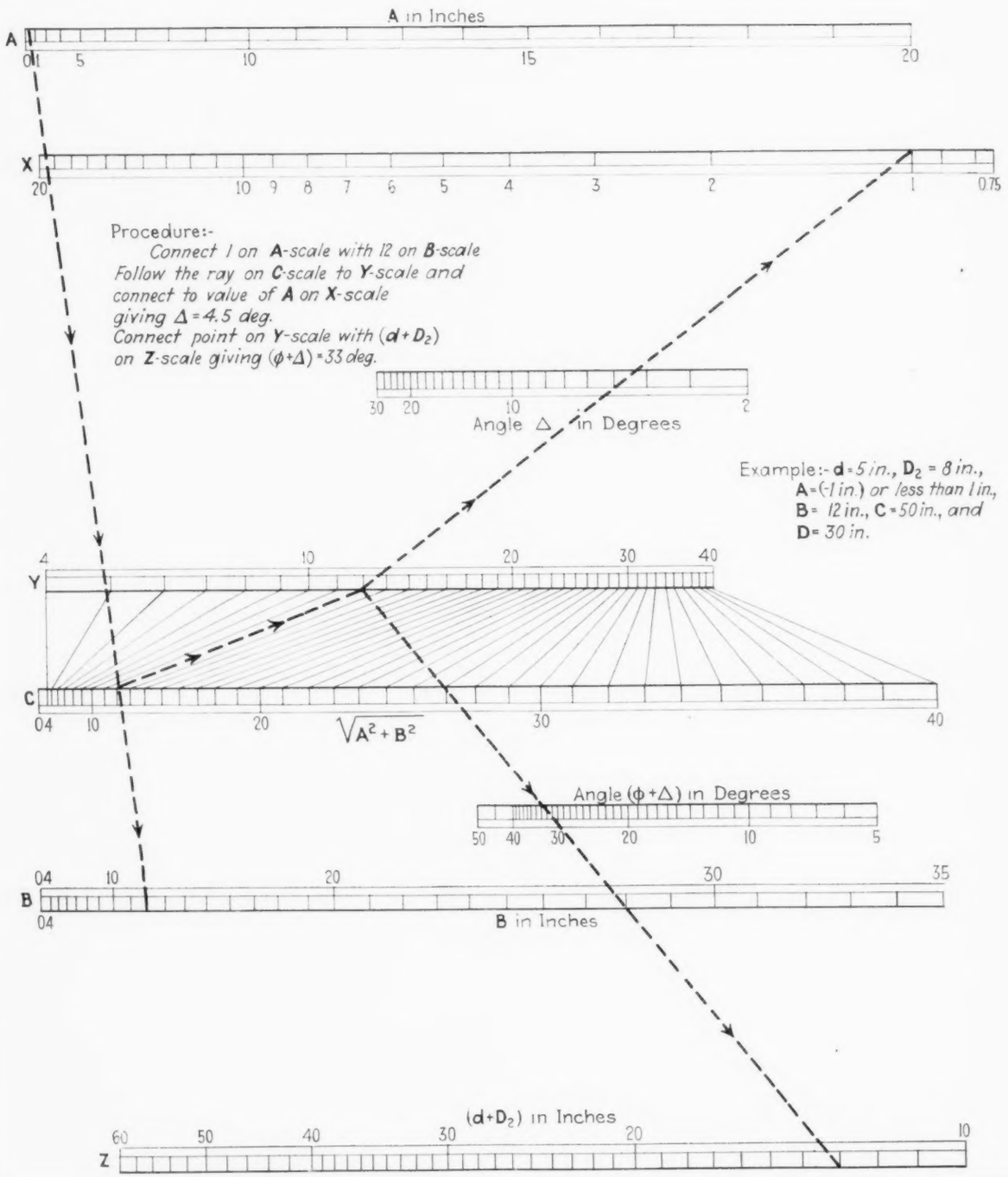


FIG. 1

Short Center Belt Drives

E. N. KEMLER
Assistant Professor of Mechanical Engineering
University of Pittsburgh


A in Inches
A
0 1 5 10 15 20
X
20 10 9 8 7 6 5 4 3 2 1 0.75
Procedure:-
Connect 1 on A-scale with 12 on B-scale
Follow the ray on C-scale to Y-scale and
connect to value of A on X-scale
giving Δ = 4.5 deg.
Connect point on Y-scale with (d + D2)
on Z-scale giving (φ+Δ) = 33 deg.
30 20 10 2
Angle Δ in Degrees
Example:- d = 5 in., D2 = 8 in.,
A = (-1 in.) or less than 1 in.,
B = 12 in., C = 50 in., and
D = 30 in.
Y
4 10 20 30 40
C
0 4 10 20 √(A² + B²) 30 40
Angle (φ+Δ) in Degrees
50 40 30 20 10 5
B
0 4 10 20 30 35
B in Inches
(d+D2) in Inches
Z
60 50 40 30 20 10

PRODUCT ENGINEERING © REFERENCE BOOK SHEET